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AN ABSTRACT OF THE THESIS OF David M. LaLiberte for the Master of Science in Civil Engineering presented August 30, 1990.

Title: Density Currents in Circular Wastewater Treatment Tanks.

APPROVED BY THE MEMBERS OF THE THESIS COMMITTEE:

Scott Wells, Chair

Roy Kock

Gerald Recktenwald

Jean Murray

Deviations from ideal flow and settling occur in circular wastewater treatment tanks because of tank geometry, flow conditions, and density currents caused by variations in suspended solids concentration and temperature distributions. Thermally induced density currents were investigated in this study. Under winter, low flow

conditions, measurements were made of vertical and radial temperature distributions in the circular chlorination tank at Lake Oswego, Or., and in the circular primary and secondary clarifiers at Bend, Or. Thermistor arrays were used to collect the data which exhibited both vertically well-mixed and a two-layer flow regime. Inlet geometry and suspended solids in the secondary clarifiers caused a warm bottom inflow and apparent thermal instability. Meteorological measurements were also made. The calculated winter heat loss values indicated that convective mixing may have inhibited particle sedimentation in the clarifiers.

A two-dimensional, radial flow model was presented which incorporated the effect of density currents due to temperature and suspended solids. The model consists of conservation of fluid mass and momentum equations expressed in cylindrical coordinates. The model equations were simplified by assuming steady-state, axisymmetric, turbulent shear flow. A set of first-order, non-linear, ordinary differential equations were obtained by depth-averaging the two layer flow equations and solved using a Runge-Kutta numerical method.

The model demonstrated many of the flow characteristics suggested in the results of the experimental studies. A thin, bottom layer inflow was simulated when values from the field studies were used for tank flow, geometry, suspended solids concentration and temperature. Sensitivity studies

demonstrated the model's applicability to a variety of cases including treatment tanks without baffles. The addition of a turbulent mixing term in the model equations would enhance the model by allowing the transport of fluid and solids across the layer interface.

DENSITY CURRENTS IN CIRCULAR WASTEWATER
TREATMENT TANKS

by

DAVID M. LaLIBERTE

A thesis submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE
in
CIVIL ENGINEERING

Portland State University
1990

TO THE OFFICE OF GRADUATE STUDIES:

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CHAPTER I

INTRODUCTION

Flow behavior in wastewater sedimentation tanks has long been known to vary from the quiescent, unidirectional flow conditions on which current designs are based. These tanks are normally sized on the basis of the surface overflow rate. The tank detention time is an important parameter for flocculent sedimentation. By neglecting the complexities of tank hydrodynamics, the predictive capabilities of ideal models are limited. Engineers are required to utilize large safety factors in tank design which demonstrates the lack of understanding of tank hydrodynamics (1,2,3,4).

Deviation of actual sedimentation tank operation from the ideal is caused by a number of factors. Significant factors include; 1) the dependence of the flow field on inlet and outlet geometry; 2) the existence of inflow jet turbulence; 3) the presence of dead zones in the tank; 4) the resuspension of settled solids; and 5) the existence of density currents induced by suspended solids (SS) concentration and temperature differences within the tank (1,2,5).

This study focuses on deviations from ideal flow and

settling that occur due to density currents, e.g., those induced by temperature differences. A critical condition under which temperature differences may prove significant is during periods of coldweather when the surface heat loss is expected to be high. When this condition is coupled with very low plant flow rates, increased detention times will result in even more heat loss (5,6). Though there are a few studies which attempt to address this specific problem, there is still very little experimental analysis of thermal effects on wastewater treatment tank operation. Under winter conditions, the thermal structure in the tank may reduce detention times by short circuiting of the inflow. Particle settling velocities may also be affected by thermally induced convective turbulence due to surface cooling.

This paper investigates water temperature distributions in various tanks at two different wastewater treatment plants. These experimental studies were carried out on the circular chlorination tank at Tryon Creek Wastewater Treatment Plant in Lake Oswego, Or., and on the primary and secondary circular clarifiers at the City of Bend Wastewater Control Plant in Bend, Or. An idealized wastewater treatment plant using circular, radial flow tanks is illustrated in Figure 1. Meteorological data were also recorded in order that surface heat loss in the tanks could be determined and correlated with tank water temperatures.

The different tanks studied provided an opportunity to

Figure 1. Idealized wastewater treatment plant utilizing circular, radial flow tanks.

view thermal effects under a wide variety of conditions. The relative effects of suspended solids concentration and temperature difference on tank hydraulics are examined. This is accomplished by a comparison of the experimental results from the chlorination tank, primary clarifier, and secondary clarifier studies. The Lake Oswego plant, with a maximum capacity of 25 million gallons per day (MGD), had greater plant flow ($Q_{\max}$) rates than the Bend site, $Q_{\max} = 15$ MGD. This allowed the effect of thermal conditions on varying inflow inertia to be investigated.

The effect of geometry on tank hydrodynamics is also demonstrated in this paper. The tank inlet baffle is shown to be important since it affects the depth and vertical position of the inflow layer. The circular plan of all the tanks studied (Figure 2) requires that horizontal flow velocities be dependent on radial distance.

Based on the literature review of hydrodynamic models and winter heat loss models and the results of the experimental study, a mathematical model is introduced. The model consists of simplified conservation of fluid mass and momentum equations. These governing equations are expressed in cylindrical coordinates (r, θ, z) assuming steady, two-layer, turbulent shear flow. In addition, the circular tank is treated as an axisymmetric problem, i.e., all terms are independent of θ , and the model equations reduce to a two-dimensional (r, z) radial flow domain (Figure 2).

Figure 2. Coordinate system of a typical circular clarifier.

After depth-averaging, the resulting first-order, non-linear, ordinary differential equations were solved using a Runge-Kutta numerical method. Sensitivity studies were performed to determine the model's response to thermal and suspended solids induced two-layer flow. These sensitivity studies were then compared to the experimental results to assess the model's applicability.

CHAPTER II

PROBLEM BACKGROUND

As proposed by Camp in 1946, the movement of suspended solids can be modeled by superimposing a particle settling velocity over a steady, uniform hydrodynamic flow pattern (7). The settling of discrete particles for an ideal circular sedimentation tank is illustrated in Figure 3A. Through the use of similar triangles, the ideal surface overflow rate, or surface settling velocity (v_o), can be expressed as

$$v_o = \frac{\bar{u}h_o}{R} = \frac{Q_o}{A_s} \quad (1)$$

Where

Q_o = tank flow rate;

$\bar{u}$ = average radial velocity;

h_o = settling depth;

R = radial flow distance;

A_s = tank surface area.

The surface overflow rate represents the velocity of the slowest settling particle that is completely removed (3,7,8). Some of those particles with a settling velocity, v_p , less than v_o will not be removed (Figure 3A). The fraction, f , of suspended solids removed can then be

Figure 3. Partial removal in sedimentation tanks of A) discrete particles; B) flocculent particles; and C) flocculent particles in shallow tanks (3).

expressed as

$$f = \frac{h}{h_o} = \frac{v_p t_o}{v_o t_o} = \frac{v_p}{(Q/A)} \quad (2)$$

Where

h = maximum height at which particle with
velocity v_p will settle out;

t_o = ideal hydraulic residence time,

$$t_o = \frac{V}{Q_o} \quad (3)$$

V = tank volume.

Note that Equations 1 and 2 imply independence of sedimentation tank efficiency from both tank depth and detention time. As a result, some tank designs have emphasized increasing the effective surface area while reducing the depth of the tank (3,7). Realistically, however, this may not be adequate since Equations 1 and 2 consider only discrete particle sedimentation in uniform flow. Implicitly, these equations assume that particles of a constant velocity are settling through the full depth of the tank. This may not be the case for flows in clarifiers which are affected by inflow inertia, density currents, and tank geometry.

The ideal settling analysis does not take into account vertical fluid velocities which may exist in the tank. Inflow turbulence, generated at the inlet pipe and baffle,

produces significant vertical fluid motion that can cause entrainment of and mixing with ambient tank water. Also, upward vertical velocities exist immediately upstream of the outlet weir (1,4). As will be shown, the particle settling velocity (v_p) is also affected by thermally generated turbulence.

Density differences, inlet conditions, and outlet conditions may produce two-layer flow which causes short circuiting of the tank inflow (2,5) and reduces detention times (Figures 4A and 5A). If a bottom layer exists, or the tank is too shallow, the potential for bottom scour due to increased velocities is significant. Scouring of particles resuspends them and effectively reduces the tank's removal efficiency.

Density differences, e.g., induced by thermal stratification and/or suspended solids (SS) concentration gradients, produces two-layer flow which affects tank operation (5). Under winter conditions, the potential of thermal short circuiting of the warm inflow exists (Figure 4B and 5B). In secondary clarifiers, the very high inflow SS concentration (~2000 mg/l) may generate a bottom density current which upwells only at the outflow weir (Figures 4C and 5C).

Discrete particle sedimentation is more indicative of primary clarifier operation. It is inappropriate to exclusively apply Equations 1 and 2 to secondary clarifiers

Figure 4. Flow behavior in wastewater treatment tanks with inlet baffles under A) typical conditions; B) warm inflow; and C) high suspended solids (SS) loading.

Figure 5. Flow behavior in wastewater treatment tanks without inlet baffles under A) typical conditions; B) warm inflow; and C) high suspended solids (SS) loading.

since, in these tanks, it is necessary to include the significance of particle flocculation. When flocculation occurs within sedimentation tanks, particle sedimentation velocities increase with flow distance because of particle agglomeration (Figure 3B). Thus, a particle entering initially at a height greater than h , as defined in Equation 2, may still be settled out. However, in a shallow basin, flocculant particles may not achieve sufficient size and settling velocity to be removed since v_p is now a function of depth and, hence, detention time (Figure 3C). Detention time, particularly for secondary clarifiers, is then an important hydraulic parameter for determining removal efficiency (3,8).

Some studies have investigated the relationship between overflow rate ($v_o=Q/A$) and sedimentation tank performance. These investigators found considerable deviation from ideal tank conditions with very little correlation between Q/A and suspended solids removal efficiency (9,10). El-Baroudi (11) carried out tracer dye studies on wastewater sedimentation tanks and found that mean detention times were significantly less than predicted by $t_o=V/Q_o$ (Eq. 3) because of turbulent mixing and inflow short circuiting. These results led many later investigators to attempt more accurate hydrodynamic models to predict sedimentation tank removal efficiencies and to provide the basis for better tank design (1,2,3).

Experimental studies have been undertaken to determine the physical processes affecting suspended solids removal efficiency in wastewater sedimentation tanks. Clements and Khattab (12,13) carried out vertical and horizontal velocity measurements in both rectangular and circular clarifiers. Their results showed that a strong recirculating flow pattern prevailed in both the rectangular and circular clarifiers (Figure 6). Rather than the uniform horizontal velocities assumed by Camp and earlier modelers (3,7), velocities were shown to vary in magnitude and direction throughout the tank.

Clements (12) and Clements and Khattab (13) also demonstrated that inlet geometry was a dominant feature affecting flow and velocity distributions within the tank. Numerous comparisons of flow behavior were carried out on a variety of rectangular and circular tanks. Changing the inlet baffle depth, or excluding the baffle altogether, caused variability in the magnitude and the direction of the recirculating flow. When the baffle was in place, a counter-clockwise flow developed as shown in Figure 6A. When the baffle was removed, a clockwise flow developed (Figure 6B). For circular clarifiers, it was also shown that altering the slope of the tank bottom, though it produced some variation in the tank flow, did not affect the dominant recirculating flow behavior.

More recently, Tay and Heinke (2) have provided

Figure 6. Recirculating flow direction in A) tanks with baffles (CCW); and B) tanks without baffles (CW).

measurements of velocity and suspended solids concentrations inside rectangular and circular clarifiers. Their velocity measurements in wastewater sedimentation tanks support previous studies showing a dominant recirculating flow. Their suspended solids, advection-dispersion model takes into account a non-uniform flow field in its formulation. The model was calibrated by field data. Verification of the model, by correlating the results of the calibrated model with additional SS measurements, demonstrated good agreement. The model is, therefore, capable of predicting SS concentrations throughout the tank assuming a non-uniform flow field.

The above discussion has identified the discrepancies that exist between the flow behavior of the ideal settling model and the flow behavior of actual settling tanks. Because of these discrepancies, many later investigators have developed more accurate hydrodynamic models on which to base predictions of sedimentation tank performance. A more complete understanding of tank hydrodynamics will lead to more efficient tank design. An overview of hydrodynamic models currently applied to clarifiers is presented in the next chapter.

CHAPTER III

HYDRODYNAMIC MODELS

A simple hydrodynamic model for rectangular clarifiers has been proposed by Ostendorf (14) which attempts to simulate the two-dimensional velocity field present in clarifiers. In this analytical, steady flow model, the inlet zone is represented as a turbulent jet, the settling zone as a uniform flow field, and the outlet zone as a converging flow field. The settling zone is taken as a region of decaying turbulence and increasing uniform velocity. The velocity field at the end of the settling zone is assumed uniform as it enters the outlet zone. This zone is simulated by a converging flow field that approaches the effluent weir (14,16). Velocity profiles generated in this way show reasonable agreement with laboratory models. In the case of sedimentation tanks, the inclusion of a turbulent intensity parameter in the model simulates the dispersion characteristics of the flow. Dispersion in actual tanks affects thermal distributions, as well as, particle settlement and resuspension. Though this model neglects the effects of wind and stratification on the flow field, it gives a fast and simple method to investigate tank hydrodynamics.

Abdel-Gawad and McCorquodale (4) and Shamber and Larock (15) have both developed numerical models which assume two-dimensional, steady, turbulent, unstratified flow in the tank. These models simulate the flow field by solving the fluid mass continuity and momentum equations. Abdel-Gawad and McCorquodale (4,17) stated these equations in a simplified differential form for a circular clarifier assuming that the flow is axisymmetric and the pressure distribution is hydrostatic ($P_z = \rho gh$), i.e.,

$$\text{Continuity: } \frac{\partial}{\partial r} (r\bar{u}) + \frac{\partial}{\partial z} (r\bar{w}) = 0 \quad (4)$$

$$\text{r-Momentum: } \bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} = -g \frac{\partial h}{\partial r} + \frac{1}{\rho} \frac{\partial \tau}{\partial z} \quad (5)$$

Where

r = horizontal radial distance;

z = vertical distance;

h = fluid depth;

$\bar{u}$ = mean radial velocity;

$\bar{w}$ = mean vertical velocity;

$\bar{\rho}$ = mean fluid density;

τ = total shear stress.

These governing non-linear, first-order, partial differential equations were reduced to a set of ordinary differential equations through the use of the strip integral method. A dominant horizontal flow direction and velocity shape functions were assumed which accounted for the effects of the bottom boundary layer, the inflow jet potential core,

and the free mixing and recirculating zone. A Runge-Kutta method was used to integrate the set of ordinary differential equations. A finite element solution was used in the withdrawal zone since the strip integral method was inapplicable to the converging flow field assumed in that region (17).

The Abdel-Gawad and McCorquodale model neglects the effect of density differences due to variations in suspended solids concentrations and in temperature. An application of the model to simulate SS removal efficiency in a sedimentation tank was carried out. By utilizing model predictions of the kinematic eddy viscosity distribution in the tank, theoretical SS removal was compared with available data. Since the model demonstrated only limited agreement with actual tank removal efficiency, the study indicated the need for an SS transport process in the model equations (4).

The Shamber and Larock (15) model was based on many of the same assumptions previously discussed in the Abdel-Gawad and McCorquodale model. However, in order to close their model, the kinetic energy-dissipation ($k-\epsilon$) turbulence model was used to determine the variation of the turbulent eddy viscosity throughout the tank. This solution required that two turbulent transport equations be utilized in addition to the mass and momentum conservation equations. The turbulent eddy viscosity (ν_t) represents the local structure of turbulence throughout the tank. Based on dimensional

analysis, the turbulent eddy viscosity (ν_t) was considered proportional to the kinetic energy of turbulence (k) and its rate of dissipation (ϵ), i.e.,

$$\nu_t = \frac{c_\mu k^2}{\epsilon} \quad (6)$$

Where c_μ is a model constant.

This analysis resulted in a closed system of five coupled, non-linear, partial differential equations which were solved numerically by the Galerkin finite element method. The model produced results similar to those of the Abdel-Gawad and McCorquodale study in that non-uniform velocities and turbulence were predicted throughout the tank (4,15). However, the k - ϵ closure model and Galerkin finite element solution require a greater number of model parameters and longer computation time. Also, this model does not attempt to simulate flow patterns where stratified conditions, or density induced turbulence, exists due to thermal and SS concentration differences.

In the development of the modified k - ϵ turbulence model for sedimentation tanks proposed by DeVantier and Larock, an effort has been made to account for suspended solids affected density currents (1,18,19). Sediment driven density currents result when a heavily laden suspended solids inflow travels as a bottom current under the less dense clarified tank fluid. [Note that DeVantier and Larock's model is based on the earlier work by Shamber and Larock (15).] DeVantier

and Larock modified the k- ϵ turbulence model proposed by Rodi (20) with the inclusion of a sediment transport equation. For steady, radial flow, the model's time-averaged mass, r-momentum, z-momentum, and sediment volume conservation equations are expressed, respectively, as

$$\frac{\partial}{\partial r} (r\bar{u}) + \frac{\partial}{\partial z} (r\bar{w}) = 0 \quad (7)$$

$$r \left(\bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} \right) = - \frac{r}{\bar{\rho}} \frac{\partial P'}{\partial r} - \frac{\partial}{\partial r} (\overline{ru'u'}) - \overline{v'v'} - \frac{\partial}{\partial z} (\overline{u'w'}) r \quad (8)$$

$$r \left(\bar{u} \frac{\partial \bar{w}}{\partial r} + \bar{w} \frac{\partial \bar{w}}{\partial z} \right) = - \frac{r}{\bar{\rho}} \frac{\partial P'}{\partial z} - \frac{\partial}{\partial z} (\overline{ru'w'}) - \eta \bar{\Lambda} g r - \frac{\partial}{\partial z} (\overline{w'w'}) r \quad (9)$$

$$r \left(\bar{u} \frac{\partial \bar{\Lambda}}{\partial r} + \bar{w} \frac{\partial \bar{\Lambda}}{\partial z} \right) = \frac{\partial}{\partial z} \left(r \bar{\Lambda} (1 - \bar{\Lambda}) v_s \right) - \frac{\partial}{\partial r} (\overline{ru'\lambda}) - r \frac{\partial}{\partial z} (\overline{w'\lambda}) \quad (10)$$

Where

r = radial distance;

z = vertical distance;

h = fluid depth;

$\bar{u}$ = mean horizontal velocity;

$\bar{w}$ = mean vertical velocity;

$\bar{\rho}$ = mean fluid density;

P' = dynamic pressure;

$\bar{\Lambda}$ = mean solids sediment volume fraction,

$$\bar{\Lambda} = \frac{\text{mean volume of solids}}{\text{total volume of mixture}} \quad (11)$$

u' = radial turbulent fluctuation;

w' = vertical turbulent fluctuation;

v' = axial turbulent fluctuation;

λ = sediment volume concentration fluctuation;

η = non-dimensional density difference between
the pure fluid (ρ_f) and pure solid density
(ρ_s),

$$\eta = \frac{\rho_s - \rho_f}{\rho_f} \quad (12)$$

g = gravitational acceleration;

$\eta \bar{\Delta} g$ = excess body force due to sediment;

v_s = suspended solids settling velocity.

The Boussinesq approximation for density affected flows was applied to the r-momentum and z-momentum equations. The approximation is based on the assumption that variation in bulk fluid density is important only in the body force term, i.e., $\eta \bar{\Delta} g$. Thus, the fluid density in all other terms can be assumed equal to that of pure water at a specific temperature.

As in the Shamber and Larock model (15), the turbulent flux terms are modeled using the eddy viscosity concept. Kinetic energy and dissipation transport equations, and the kinematic eddy viscosity, $\nu_t = c_\mu k^2 / \epsilon$, are introduced to close the model. The additional transport equations are utilized in a modified form of the k- ϵ turbulence model and can be expressed as (1,20)

$$\bar{u} \frac{\partial k}{\partial r} + \bar{w} \frac{\partial k}{\partial z} = \text{Pr} - \epsilon + \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\nu_t}{\sigma_k} \frac{\partial k}{\partial r} \right) + \frac{\partial}{\partial z} \left(\frac{\nu_t}{\sigma_k} \frac{\partial k}{\partial z} \right) \quad (13)$$

$$\begin{aligned} \bar{u} \frac{\partial \epsilon}{\partial r} + \bar{w} \frac{\partial \epsilon}{\partial z} = & c_{\epsilon 1} \frac{\epsilon}{k} \left(\text{Pr} + (1 + c_{\epsilon 3}) B \right) - c_{\epsilon 2} \frac{\epsilon^2}{k} \\ & + \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\nu_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial r} \right) + \frac{\partial}{\partial z} \left(\frac{\nu_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial z} \right) \end{aligned} \quad (14)$$

Where

Pr = production of turbulent kinetic energy by mean shear,

$$\text{Pr} = \nu_t \left[2 \left(\frac{\partial u}{\partial r} \right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial r} \right)^2 + 2 \left(\frac{\partial w}{\partial z} \right)^2 + 2 \left(\frac{u}{r} \right)^2 \right] \quad (15)$$

B = production of turbulent kinetic energy by buoyancy,

$$B = \eta g E_s \frac{\partial \bar{\lambda}}{\partial z} \quad (16)$$

E_s = turbulent diffusion coefficient ($E_s \approx 1.2 \nu_t$).

The values of the model constants are (20) $c_{\epsilon 1} = 1.44$; $c_{\epsilon 2} = 1.92$; $c_{\epsilon 3} = 0.8$; $c_\mu = 0.09$; $\sigma_\epsilon = 1.3$; $\sigma_k = 1.0$. Solution of this set of six partial differential equations (Equations 7-10,13,14) requires that all inlet, outlet, water surface, tank wall and tank bottom boundary conditions be known.

In the final model formulation by DeVantier and Larock, the buoyancy production term, B, is neglected because of computational limitations and because the effect of buoyancy

on tank hydrodynamics was considered of secondary importance. In the first case, the inclusion of the buoyancy term caused instability of the Galerkin finite element method. When included in the model, buoyancy acted as a net sink of turbulent kinetic energy (k) which resulted in large negative values for k and ϵ . This condition was considered physically impossible in the operation of sedimentation tanks (1,21).

In the second case, neglecting buoyancy as a minor component of tank hydrodynamics was based on a consideration of the flux Richardson number (Ri)

$$Ri = \frac{B}{Pr} \quad (17)$$

As noted by Turner (21), the flux Richardson number (Ri) has a maximum range of 0.1 to 0.3 above which turbulence collapses and buoyancy damping can be assumed to have only secondary effect upon the values of k and ϵ . However, Wells (22) has pointed out that neglecting buoyancy may not be appropriate under cold weather conditions when the effect of buoyancy production on turbulence generation can be significant. In this case, the flux Richardson number can be shown to be less than -1 which implies that turbulent production is dominated by surface cooling rather than by mean shear (5).

Although the DeVantier and Larock model neglects the effect of buoyancy induced turbulence in tank hydrodynamics,

the model is still capable of relating local eddy viscosity to flow characteristics dependent on density currents. This makes the model applicable to secondary clarifiers where the density gradients are a significant factor affecting flow. Unfortunately, no direct experimental verification of the model yet exists. This model may also be improved by modifying the governing equations to include a heat transport equation. With surface heat flux boundary conditions, the model could then simulate thermal effects on flow behavior, and sedimentation, in treatment tanks.

CHAPTER IV

WINTER HEAT LOSS MODELS

In lieu of a comprehensive model for sedimentation tank hydrodynamics including buoyancy effects due to temperature, some investigators have developed temperature models to predict temperatures and thermal conditions in these tanks (5,6). Methods which parameterize the significance of thermally induced stratification and convective mixing are available in the literature (5,21,23). Thermal instability generated by surface cooling can be substantial during winter and low flow conditions for uncovered treatment tanks (5,6). [Note that biological and chemical processes are also dependent on tank temperature conditions (8). However, a thorough investigation of these processes is beyond the scope of this paper and the present discussion will focus on temperature affected tank hydrodynamics.]

Wall and Petersen (6) developed a heat transfer model intended to predict the wintertime equilibrium temperature of exposed wastewater clarifiers or any open tank. This model was composed of five heat transfer terms. These include 1) convection 2) evaporation 3) radiation from liquid to air 4) solar irradiation and 5) heat supplied by influent water.

The equilibrium water temperature utilized in their model was determined by equating the heat loss and the heat gain. Expressed as a heat balance,

$$Q_{\text{lost}} = Q_{\text{gain}}$$

$$Q_{\text{nc}} + Q_{\text{fc}} + Q_{\text{e}} + Q_{\text{r}} = Q_{\text{s}} + Q_{\text{iw}} \quad (18)$$

Where

Q_{nc} = natural convection;

Q_{fc} = forced convection (dependent on wind conditions);

Q_{e} = evaporative heat loss;

Q_{r} = heat radiation from water to air;

Q_{s} = solar irradiation (short wave) heat transfer;

Q_{iw} = inflow heat transfer.

Though Equation 18 is a convenient analytical expression, it implicitly assumes well-mixed conditions in the tank (23). It has been shown previously that stratification may exist in many wastewater treatment tanks. While most effort has focused on sediment induced stratification, thermally induced stratification may also exist under winter conditions, particularly, when tank flow rates are low (5,6).

Refinements in the meteorological dependent surface heat flux terms can also be made (23). A heat term representing atmospheric radiation, primarily due to the presence of water vapor in the atmosphere, may be significant and should be included. Air temperature and wind

speed data collection should be taken at known elevations in order to account for the effects of fetch (i.e., boundary layer effects.) Measuring the relative humidity throughout the day would more accurately reflect changing meteorological conditions. Finally, the frequency of data collection should be on an hourly basis, rather than daily, since the tank detention times are of that order.

Whether thermally induced two-layer flow exists can be determined by an analysis of the Pond number ($\mathcal{P}$). As expressed by Jirka (24),

$$\mathcal{P} = \frac{h_s}{H} = \left[\frac{f_i}{4} \frac{Q^2}{\beta \Delta T g H^3 B^2} D_v^3 \frac{L}{H} \right]^{1/4} \quad (19)$$

Where

h_s = surface layer depth;

f_i = interfacial friction factor (0.01- 0.001);

Q = flow rate;

β = coefficient of thermal expansion of water;

ΔT = total temperature difference across tank;

D_v = vertical entrance dilution;

g = acceleration due to gravity;

L = length of tank;

B = width of the tank;

H = total depth of the tank.

The Pond number parameterizes the relative effect of the destabilizing turbulent inflow energy and the stabilizing effects of thermally induced stratification.

Jirka has shown that, for $\mathcal{P} < 0.3$, stratification exists, and for values for $\mathcal{P} > 1.0$, a well-mixed condition exists. However, this formulation is limited to rectangular tanks where the surface layer velocity and depth are constant. It also neglects density effects because of suspended solids.

Based on the above considerations and refinements, Wells (5) has proposed a more comprehensive temperature model for wastewater tanks that evaluates two-layer flow caused by thermal conditions. The temperature transport equation was assumed to govern the temperature distribution in the tank

$$\frac{\partial T}{\partial t} = D_L B^2 \frac{\partial^2 T}{\partial A^2} - \frac{Q}{H} \frac{\partial T}{\partial A} + \frac{\phi_n}{\rho c_p h} \quad (20)$$

Where

T = temperature;

t = time;

D_L = longitudinal dispersion coefficient;

B = width of tank;

h = surface layer depth;

A = surface area;

H = total flow depth;

c_p = specific heat of water at a constant pressure;

ϕ_n = net surface heat flux.

The net surface heat flux (25), ϕ_n , was determined from

$$\phi_n = \phi_{sn} + \phi_{an} - (\phi_{br} + \phi_e + \phi_c) \quad (21)$$

Where

ϕ_{sn} = net solar radiation (short wave);

ϕ_{an} = net atmospheric radiation (long wave);

ϕ_{br} = back radiation of water (long wave);

ϕ_e = evaporative heat flux;

ϕ_c = convective heat flux.

The net radiation term, $\phi_r = \phi_{sn} + \phi_{an}$, is dependent on meteorological conditions (25) and can be expressed in BTU/(ft²·d) as

$$\phi_r = \phi_{sn} + 1.16 \times 10^{-13} (460 + T_a)^6 (1 + 0.17C^2) \quad (22)$$

Where

T_a = air temperature (°F);

C = cloud cover, where $0 \leq C \leq 1$ with

$C=0$ representing clear sky;

The total heat loss (ϕ_L)

$$\phi_L = \phi_{br} + \phi_e + \phi_c \quad (23)$$

is dependent on surface water temperatures as well as meteorological conditions (25) and can be expressed in BTU/(ft²·d) as

$$\phi_L = 0.97\sigma(T_s + 460)^4 + f(w) \left[(e_s - e_a) + c_b(T_s - T_a) \right] \quad (24)$$

Where

T_s = water surface temperature (°F);

$f(w)$ = wind speed function for heat flux;

$$f(w) = 22.4(\Delta\theta_v)^{1/3} + 14.0w_2;$$

w_2 = wind speed at 6 ft (~2m) above ground level;

e_s = saturated vapor pressure at temperature of
water surface (mm Hg);

$$e_s = 25.4 \exp(17.62 - 9500.8 / (T_s + 460));$$

e_a = saturated vapor pressure at temperature of
air measured at 6 ft above ground (mm Hg);

$$e_a = 25.4 \exp(17.62 - 9500.8 / (T_a + 460));$$

c_b = conduction heat flux coefficient;

$$c_b = 0.255 \text{ mm Hg}/^{\circ}\text{F} \quad (0.61 \text{ mb}/^{\circ}\text{C});$$

σ = Stefan-Boltzman constant;

$$\sigma = 4.15 \times 10^{-8} \text{ BTU}/(\text{ft}^2 \cdot \text{d} \cdot ^{\circ}\text{R}^4);$$

$$\Delta\theta_v = T_{sv} - T_{av};$$

$$T_{sv} = [(T_s + 460) / (1 - 0.378 e_s / P_a) - 460];$$

$$T_{av} = [(T_a + 460) / (1 - 0.378 e_a / P_a) - 460];$$

P_a = atmospheric pressure.

By linearizing the temperature transport equation (Eq. 20), an analytical solution can be developed (5,26). This linearization procedure is accomplished by introducing the concept of the equilibrium temperature, T_E , at which the surface heat flux equals zero (Figure 7). By assuming $\phi_n = 0$ in Equation 21, i.e., no net heat transfer across the air/water interface, the surface water temperature at which equilibrium occurs for a specific set of weather conditions can be determined (25).

An iterative method for solving Equation 21 implicitly

Figure 7. Plot of ϕ_n Vs. T_s using the equilibrium temperature concept.

for T_E is usually employed. Initially, under specific meteorological conditions, a value for T_E is chosen and used as an initial guess in Equation 21. If the resulting value of ϕ_n is not equal to zero, the process is continued until a value for T_s is chosen such that $\phi_n=0$.

Utilizing the equilibrium temperature concept, the surface heat exchange coefficient, K , can be defined as the slope of the line relating the surface heat flux, ϕ_n , to the water surface temperature, T_s . This linear relationship is illustrated in Figure 7 and can be expressed as

$$K = -\phi_n / (T_s - T_E) \quad (25)$$

Where

T_s = temperature of surface water;

T_E = equilibrium temperature of water;

K = surface heat exchange coefficient.

The surface heat exchange coefficient represents the rate of surface heating or cooling under specified meteorological conditions.

Assuming steady-state conditions, neglecting dispersion, and utilizing the above approximation (Eq. 25), the temperature equation (Eq. 20) becomes

$$\frac{Q}{h} \frac{dT}{dA} = \frac{-K}{\rho c_p h_s} (T - T_E) \quad (26)$$

If $T=T_o$ at $A=0$, then a solution to this equation can be shown to be

$$\frac{T - T_E}{T_O - T_E} = \exp(-r) \quad (27)$$

Where

T_O = inflow water temperature

$$r = KA/\rho c_p Q \quad (27a)$$

Thus, the exit water temperature can be determined if r , T_O , and T_E are known. The time scale, or relaxation time, for a given body of water to reach T_E is expressed as

$$t_r = \rho c_p h_s / K \quad (28)$$

The Richardson flux number, R_f , can be derived once the net surface heat flux is determined (5,21). This parameter represents the ratio of turbulent energy absorbed by buoyancy to that generated by shear

$$R_f = \frac{-\alpha \beta h}{u_*^3} \quad (29)$$

Where

α = Von Karman constant (≈ 0.4);

β = buoyancy flux,

$$\beta = \frac{-\beta g}{\rho c_v} \phi_n \quad (30)$$

c_v = specific heat of water at constant volume;

u_* = shear velocity;

$$u_* = \bar{u} \left(\frac{f_o}{8} \right)^{1/2} \quad (31)$$

f_o = friction factor;

$\bar{u}$ = average horizontal velocity.

Where $R_f < 0$, buoyancy induced turbulence in the tank is significant compared to that produced by shear. In cases where $R_f \ll -1$, convective mixing dominates and the buoyancy related turbulent velocity scale, u_t , can be used to indicate the order of magnitude of thermally induced vertical mixing

$$u_t \sim (Rh)^{1/3} \quad (32)$$

If $u_t \geq v_o$ where $v_o = Q_o/A_s$ (Eq. 1), then particle sedimentation in the tank could be significantly affected.

Very little field data was available in the literature pertaining to the magnitude of thermal stratification or vertical convective mixing in sedimentation tanks. For this reason, it was necessary to conduct an experimental study to investigate the thermal parameters discussed in this chapter.

CHAPTER V

EXPERIMENTAL SET-UP

Three tanks at two different wastewater treatment plants were chosen for measurement of the vertical and radial temperature distributions. One study was made of the circular chlorination tank at the Tryon Creek Wastewater Treatment Plant in Lake Oswego, OR. Two studies, over two different time periods, were carried out for both the primary and secondary circular clarifiers at the City of Bend Wastewater Control Plant in Bend, OR. The general dimensions shown in Figure 8 are given specific values in Table I for each study tank.

For convenience, the Lake Oswego study is referred to as OSW, the first Bend study as BENDI, and the second Bend study as BENDII. In addition to the water temperature data, meteorological data were also taken at each site during each study period. With known meteorological conditions, determination of the tank's surface heat flux (ϕ_n) was possible.

Measurement of the temperature distributions for all three tanks were performed using the same equipment and techniques. The thermistors used (YSI Model 44202) in the experiment had a measurement range between -5°C and 45°C with an accuracy of $\pm 0.1^{\circ}\text{C}$ as provided by the manufacturer.

Figure 8. General dimensions for circular tanks.

TABLE I
TANK DIMENSIONS AT THE LAKE OSWEGO
AND BEND SITES

TANK DIMENSIONS	<u>OSW</u> CHLORINATION TANK	<u>BEND</u> PRIMARY CLARIFIER	<u>BEND</u> SECONDARY CLARIFIER
R (ft)	37.5	32.5	40.0
r_b (ft)	4.0	7.0	7.0
H_b (ft)	6.0	8.0	8.0
H_{max} (ft)	9.9	11.5	15.0
H_{min} (ft)	8.9	10.5	11.5
$\bar{H}_e$ (ft)	9.4	9.3	10.0
A_{surf} (ft ²) [m ²]	4420 [410]	3320 [308]	5027 [467]
Tank (ft ³) Volume [MGAL]	41,500 [0.31]	36,500 [0.27]	67,000 [0.50]
Bottom Slope	1:40	1:32	1:12

The thermistors were attached to a 10 ft long, 1 inch diameter PVC pipe and were spaced to allow 4 to 5 thermistors to be placed as an array along the length of the pipe (Figure 9). A total of three thermistor arrays were positioned radially along the tank service walkway (Figure 10). The radial and vertical position of each of the thermistors is shown in Table II. Axial temperature variations were not considered in the experimental set-up since earlier studies showed only small hydrodynamic variation in the axial direction (1,2,13).

Figure 9. Thermistor array showing depths of individual thermistors.

Figure 10. Thermistor array placement.

TABLE II

RADIAL POSITION OF EACH THERMISTOR ARRAY
AND DEPTH OF EACH THERMISTOR

THERM. POSITION	OSW CHLORINE CONTACT	BENDI PRIMARY CLARIFIER	BENDI SECONDARY CLARIFIER	BENDII PRIMARY CLARIFIER	BENDII SECONDARY CLARIFIER
r_{t1} (ft)	5.0	9.0	13.0	9.0	13.0
r_{t2} (ft)	18.0	19.0	25.0	19.0	25.0
r_{t3} (ft)	33.0	30.0	37.0	30.0	37.0
z_1 (ft)	0.25	0.50	0.25	0.25	0.25
z_2 (ft)	1.75	3.00	1.00	1.75	1.00
z_3 (ft)	3.75	5.50	3.00	3.75	3.00
z_4 (ft)	6.25	7.50	5.50	6.25	5.50
z_5 (ft)	8.75	—	—	8.75	8.25

A relative calibration of the YSI thermistors was carried out in a water bath using three different reference temperatures. All of the sixteen thermistors used in the experiment were loosely bundled together so that each thermistor would hang without touching each other, the bottom or the sides of the insulated container during calibration. Data logging was started after temperatures in the bath were stable for a few minutes. The data logger recorded the temperature vs. time for each thermistor and a calibration reference temperature was measured using a mercury thermometer. The thermometer had a total temperature range from 0°C to 40°C with a resolution of 0.1°C.

By comparing the logged temperature values of the thermistor to the known water temperature as measured by the mercury thermometer, the error temperature difference, $\pm\Delta T$, could be determined. Since the voltage/temperature relation for the thermistors was linear over the calibrated range (Appendix A), and because the temperature difference varied about the same amount for each of the three calibration temperatures, a linear average was used for correction to the thermistor temperature scale.

Calibration was completed by editing the display command file, DCF (Appendix B), provided in the logger support software (27). The calibrated thermistors were verified once in the lab and once in the field for each study period and recalibrated if necessary. Fortunately, due to the reliability of the thermistors, recalibration was required infrequently. Finally, with an error of $\pm 0.1^{\circ}\text{C}$ introduced by thermistor accuracy and another $\pm 0.1^{\circ}\text{C}$ introduced by the calibration technique, the total error in measurement was calculated as $\pm 0.2^{\circ}\text{C}$.

One major difference in measuring temperature profiles in the Bend sedimentation tanks, as opposed to the Lake Oswego chlorination tank, was that the thermistor probes had to be removed from the tanks periodically to allow the rotating rake and skimmer arms to pass (Figure 2). The single skimmer arm was necessary for surface scum removal, while the two, oppositely placed, rake arms were necessary

for bottom sludge collection. Both the surface and bottom arms affected the length of time the probes could be left in position. In particular, when the thermistor probes were extended deep into the tank, as in the BENDII study, the probes would contact the rake arm structure. Furthermore, during these studies the sludge layer was distorted as the rake arm approached. This had the effect of pushing the probes in the direction away from the advancing rake arms. Temperature readings were then recorded for about every 20 minutes between sweeps of the rake and skimmer arms. The removal and replacement of the thermistor probes was done in about 2 minutes and a reasonably continuous temperature profile was recorded for the 2-3 hour study periods.

In the Bend studies, it was necessary to determine the depth of the sludge layer surface. This task was accomplished by taking soundings with a length of PVC pipe. The pipe, which had gradations indicating depth, was lowered down into the water until it contacted the sludge surface layer. One difficulty was determining exactly how much bottom resistance constituted 'contact' since sludge layers in sedimentation tanks have no well defined, abrupt surface. However, a consistent resistance was estimated each time and the sludge layer surface in each of the tanks was determined.

All of the tanks studied were circular and uncovered, being exposed to weather conditions. In order to determine

the effects of thermally induced stratification, or convective turbulence, it was necessary to choose a study period coinciding with winter weather conditions when surface heat loss would be significant (3). These conditions were most likely to occur during early morning hours between 12 midnight (MN) and 8 AM when plant flow rates were small and winter cooling was greatest.

A meteorological station was set up at each site for each study period. The station measurement equipment (Unidata Model 6504d) was positioned on the end of a 6 ft long galvanized iron pole that was supported by a tripod anchored to the ground. The meteorological station measured air temperature, relative humidity, barometric pressure, solar radiation, wind direction and wind speed. All of these parameters were required for determining the surface heat flux of the tank. The meteorological display command file (DCF), presented in Appendix C, shows the logging instructions for all the meteorological instruments.

A simple schematic of the electronic data logging components can be seen in Figure 11. Connection of the meteorological instrument cluster to the input channels is made via a 25 pin 'D' socket at the top of the data logger. The input channels are capable of receiving analog voltage, pulse counting, digital and serial input signals. The input channel used depends on the type of measurement instrument connected and the input signal required (27).

Figure 11. Schematic of the electronic data logging components.

The battery operated data logger (Unidata Model 6003A) was programable and was able to store recorded data in a 64k byte RAM memory. The total logging time was dependent on the number of instruments attached and the frequency of measurement. A serial computer interface (RS-232 cable) was used to transfer information to and from a host computer. In the field, a portable computer (Toshiba 1000T) was used for loading and downloading the data logger (Figure 11).

The thermistors required a high resolution (HIRES) analog interface (UNIDATA Model 6006A) in addition to the data gathering hardware already outlined above (Figure 11). The HIRES interface allowed the data logger to record input signals requiring greater accuracy and a larger dynamic signal range. Without the HIRES interface, analog voltage inputs were logged with 8 bit resolution and the analog signal range was from 0 to 2.5 volts. With the HIRES interface, the analog signal range was boosted to between -2.5 and +2.5 volts and measurement accuracy was enhanced by an analog voltage input resolution of 13 bits (27).

As stated earlier, the data logger was programable and capable of communicating with a host computer. This allowed the data logger to be programed by a set of instructions, known as a 'scheme', which defines; 1) how the logger is to operate, i.e., which channels to log, how long, how often, etc.; 2) how the logger is initialized for loading; 3) how the recorded information is retrieved for unloading; 4) how

the data will be interpreted and displayed. These instructions can be seen in the display command file in Appendix B and C. The Starlog software allows the user to create all the necessary data logging communication files, and instructions, by a menu driven program which generates the scheme required (27). A flow chart demonstrating how the thermistor data was processed is shown in Figure 12.

With the data provided from this experimental study the thermal parameters discussed in the previous chapter were determined in the next chapter.

Figure 12. Flow chart demonstrating processing of acquired data.

CHAPTER VI

RESULTS: TANK HYDRAULIC AND THERMAL PARAMETERS

Plant flow rates were provided by the treatment plant control room and are shown in Figures 13-15. The tank geometry is presented in Table I. The theoretical detention times (t_o) and overflow rates (v_o) were determined for all three study tanks (Table III).

TABLE III
IDEAL HYDRAULIC PARAMETERS FOR OSW,
BENDI AND BENDII

	OSW CHLORINE CONTACT	BENDI PRIMARY CLARIFIER	BENDI SECONDARY CLARIFIER	BENDII PRIMARY CLARIFIER	BENDII SECONDARY CLARIFIER
$\bar{Q}_o$ (cfs) [MGD]	3.63 [2.35]	0.93 [0.60]	1.32 [†] [0.85]	1.90 [1.25]	0.79 [†] [0.68]
v_o (ft/s) [cm/s]	8.0E-4 [0.024]	2.8E-4 [0.0085]	2.6E-4 [0.0080]	5.8E-4 [0.018]	2.1E-4 [0.0064]
t_o (h) [d]	3.00 [0.125]	11.0 [0.46]	14.1 [0.59]	5.30 [0.22]	17.0 [0.71]

† Plant flow rate is split between 2 secondary clarifiers.

Though flow rates through the plant varied diurnally at all the sites, the Bend studies covered a 2 to 3 hour duration which corresponded to a period of low flow through the plant (Figures 13 and 15). For the Lake Oswego study, though the study period covered about 3 days, low flow rates

Figure 13. BENDI: plant flow rates from 1-21-89 to 1-22-89.

Figure 14. OSW: plant flow rates from 2-3-89 to 2-7-89.

Figure 15. BENDII: plant flow rates for 3-4-89 to 3-5-89.

were also chosen for the analysis since it was expected that this would be a critical condition for increased surface heat loss due to the increased detention time.

In the Bend primary and secondary clarifiers, the presence of the sludge layer reduced the effective flow depth of the tank. Results of the soundings, used to determine the depth of the sludge layer, $\bar{H}_e$, and the average sludge layer thickness, $\bar{H}_s$, are shown in Table IV.

TABLE IV
EFFECTIVE FLOW DEPTHS, VOLUMES AND DETENTION
TIMES FOR OSW, BENDI AND BENDII

	OSW CHLORINE CONTACT	BENDI PRIMARY CLARIFIER	BENDI SECONDARY CLARIFIER	BENDII PRIMARY CLARIFIER	BENDII SECONDARY CLARIFIER
$\bar{H}_s$ (ft)	0	1.8	3.3	1.8	3.3
$\bar{H}_e$ (ft) [m]	9.4 [2.9]	9.3 [2.8]	10.0 [3.1]	9.3 [2.8]	10.0 [3.1]
V_e (ft ³) [MGAL]	41,500 [0.31]	30,900 [0.23]	50,300 [0.38]	30,900 [0.23]	50,300 [0.38]
t_e (h) [d]	3.00 [0.125]	9.3 [0.39]	10.6 [0.44]	4.60 [0.19]	13.4 [0.56]
% reduct. in t_o	0.0	6.8	24.9	12.2	21.0

The greater sludge layer thickness for the secondary (3.3 ft) was not surprising due to the greater inflow suspended solids concentration (~2000 mg/l) as compared to the primary (~200 mg/l). Once the effective flow depth ($\bar{H}_e$) was determined (Figure 8), the effective volume, V_e , could

be calculated. The effect of this reduced volume was to reduce actual (t_e) as compared to ideal detention times (t_0). This was demonstrated by the secondary clarifier which showed a decrease in detention time of over 20% (Table IV).

The height of the sludge layer also affected the inflow depth since it reduced the amount of clearance under the baffle. As will be demonstrated later, determining the flow depth at the baffle is an important physical parameter because, in many cases, it may dominate the dynamics of the inflow layer for a considerable radial distance from the centerwell. As indicated in Figure 4, when a baffle is in place, an inflow bottom layer is induced. When there is no baffle, or the buoyancy induced by temperature differences is great enough, a warm inflow may enter the tank as a surface flow (Figure 5) and cause short-circuiting. The presence of two-layer flow may further reduce detention times.

The Bend studies showed that a two-layer flow system existed in the secondary clarifier. Consistently, the surface layer was cooler than the lower layer. The bottom inflow layer, which was about 1°C higher than the surface temperature, had a depth on the order of H_{baffle} (Table I).

After reviewing the experimental data, numerous problems existed in attempting to apply the Pond number criterion as presented by Jirka (24) to wastewater treatment tanks. These problems include: 1) Jirka takes the surface layer as the layer of inflow and assumes that $h_1 \ll h_2$. This

is not the case for treatment tanks with baffles which have a bottom inflow layer and have a flow depth proportional to the surface layer depth. 2) Jirka's analysis is only applicable to rectangular water bodies and assumes that the surface layer flow depth and velocity are constant. For circular tanks, the flow depth and velocity may vary with radial distance. 3) Jirka's analysis does not consider density affects other than those caused by changes in water temperature. Hence, a more detailed analysis than that shown by Jirka is required to model adequately thermal effects in wastewater treatment tanks. Such a detailed analysis must take into account changing flow depths and velocities, density changes related to suspended solids, and the effect of a bottom inflow layer. This analysis is demonstrated later in the development of the two layer flow model.

The Richardson flux number, R_f (Eq. 29), describes the nature of turbulence in the wastewater treatment tanks. Using the meteorological data and recorded tank temperatures (Table V), the net surface heat flux (ϕ_n) was determined from Equation 21 for each of the sites. Winter night conditions, with a net solar radiation (ϕ_{sn}) equal to zero, coupled with low air temperatures produced large negative values for ϕ_n (Table VI). This indicated that considerable heat loss was occurring during the study periods. Heat loss of the magnitude calculated in Table VI also implied that significant convective mixing was present.

Calculation of the buoyancy flux (R) by Equation 30 and the shear velocity (u_*) by Equation 31 were performed as shown in Appendix D. The Richardson flux number (R_f) was then determined from Equation 29 (see Appendix D). R_f indicated the magnitude of buoyancy induced turbulence relative to the shear induced turbulence. The results are presented in Table VI. Since R_f for all the experimental cases was very large and negative, turbulence induced by surface cooling was the predominate source of turbulence in the tank.

TABLE V
RECORDED METEOROLOGICAL DATA FOR OSW,
BENDI AND BENDII

	OSW CHLORINE CONTACT	BENDI PRIMARY CLARIFIER	BENDI SECONDARY CLARIFIER	BENDII PRIMARY CLARIFIER	BENDII SECONDARY CLARIFIER
$\bar{T}_a$ ($^{\circ}\text{F}$) [$^{\circ}\text{C}$]	14.5 [-9.7]	37.0 [2.8]	28.0 [-2.2]	19.0 [-7.2]	37.0 [2.8]
$\bar{T}_s$ ($^{\circ}\text{F}$) [$^{\circ}\text{C}$]	51.8 [11.0]	57.6 [14.3]	56.7 [13.7]	55.4 [13.0]	54.5 [12.5]
$\bar{W}_s$ (mph)	0.5	0.0	3.0	0.0	2.5
$\bar{W}_d$	270 $^{\circ}$ (W)	————	270 $^{\circ}$ (W)	————	0 $^{\circ}$ (N)
$\overline{\text{RH}}$ (%)	60	100	50	90	90
$\bar{P}_a$ (psi)	15.0	13.9	13.9	13.9	13.9
C	0.5	0.5	0.5	0.5	0.5

TABLE VI
THERMAL PARAMETERS FOR OSW, BENDI AND BENDII

	OSW CHLORINE CONTACT	BENDI PRIMARY CLARIFIER	BENDI SECONDARY CLARIFIER	BENDII PRIMARY CLARIFIER	BENDII SECONDARY CLARIFIER
$\bar{u}$ (ft/s) [cm/s]	0.0085 [0.026]	0.0014 [0.043]	0.0018 [0.055]	0.0029 [0.088]	0.0014 [0.043]
u_* (ft/s) [cm/s]	0.0003 [0.009]	0.00005 [0.0015]	0.00006 [0.0019]	0.0001 [0.003]	0.00005 [0.0015]
BTU/ft ² /d ϕ_n [W/m ²]	-645 [-84.4]	-635 [-83.3]	-794 [-104]	-648 [-85.2]	-645 [-84.8]
R cm ² /s ³	0.00020	0.00020	0.00024	0.00020	0.00020
R_f	-3.1e+4	-6.8e+6	-4.2e+6	-8.4e+5	-7.2e+6
u_t (ft/s) [cm/s]	0.013 [0.39]	0.013 [0.39]	0.014 [0.42]	0.013 [0.38]	0.013 [0.38]
$\bar{T}_E$ (°F) [°C]	-22.8 [-30.4]	-10.3 [-23.5]	17.5 [-8.1]	-22.8 [-30.4]	16.0 [-8.9]
$K^\dagger$	8.69 [2.05]	9.36 [2.21]	20.20 [4.77]	8.30 [1.96]	16.8 [3.96]
t_r (d)	67.7	62.1	31.0	70.0	37.3

† Units are in BTU/ft²/d/°F [W/m²/°C]

Because $R_f \ll -1$, the turbulent velocity scale, u_t (Eq. 32), was appropriate for estimating the magnitude of vertical convective mixing (see Appendix D). A comparison of values for u_t in Table VI with v_o in Table III shows that u_t is significantly greater than v_o by approximately two orders of magnitude. Thus, under conditions experienced in the

field studies, convective mixing inhibited particle settling.

The equilibrium temperature (T_E) was also calculated for each of the study sites under the recorded meteorological conditions. Since the study was carried out at night ($\phi_s=0$), with air temperatures near or below freezing (Table V), equilibrium temperatures were very low (Table VI). The surface heat exchange coefficient (K), calculated by Equation 25, was determined based on the recorded meteorological data (see Appendix E).

The relaxation time, t_r , was determined from Equation 28 using the full depth of the tank. The calculated values demonstrate that it is unlikely that tank water temperatures will approach the equilibrium temperature since $t_r \gg t_0$ (Tables III and VI). If a two layer system is assumed, then t_r will be reduced since only the depth of the surface layer is affected by surface cooling, i.e., $h_s < \bar{H}_e$.

CHAPTER VII

RESULTS: VERTICAL WATER TEMPERATURE PROFILES

Under winter conditions using temperature as a tracer, the warmer inflow water can be distinguished from the colder, ambient tank water. The presence of the baffle in all three studies caused the influent to pass into the tank as a bottom layer (1,4). This was verified in many of the studies since higher water temperatures were recorded at the deepest thermistors. Though tank geometry, tank operating conditions, and meteorological conditions varied slightly for each of the sites, this apparent thermal instability was consistent between most of the tanks. Later, in analyzing the results of the model simulation, it will be shown that this apparent instability may be of only secondary importance when inflow inertia and SS are significant. This is particularly true immediately downstream of the baffle.

BENDI

During BENDI primary clarifier study, most water temperatures ranged between 14.6°C and 14.2°C (Figures 16A-C and 17C) indicating well-mixed conditions throughout a deep surface layer. However, at a radial distance of 19 ft and a depth of 7.5 ft (deepest thermistor) the temperature just

Figure 16. BENDI primary vertical temperature profiles from 1-21-89 (5:30AM-7:30AM) at radial distance of: A) 9 ft; B) 19 ft; C) 30 ft.

Figure 17. BENDI primary from 1-21-89 (5:30AM -7:30AM): A) air temperatures; B) plant inflow rates C) vertical water temperatures (R=19 ft).

before 6 AM suddenly increased (Figures 16B and 17C). The average temperature for this probe was 14.7°C over the period after 6 AM. Although the average temperature difference of 0.4°C , between 14.6°C and 14.2°C , is on the same order as the error introduced by the thermistors ($\pm 0.2^{\circ}\text{C}$), the apparent warm, upwelling current is distinctive and persistent. It can be seen in Figure 16B that increased bottom layer temperatures remained for over an hour and a half.

In general, water temperatures went down from 14.4°C to 14.1°C over the 2 hour period (Figures 16A-C). During this period, air temperatures were seen to increase about 3.5°C , from 0.5°C to 4.0°C , before falling back down to 0°C (Figure 17A). This indicated that the tank surface layer showed little response to short term changes in air temperature and other meteorological conditions (see Figure 17C, depth= 0.5 ft). This study period corresponded to conditions of very low flow, $Q_0 \approx 0.93$ cfs (0.6 MGD) (Figure 17B).

The temperature data recorded for the BENDI secondary clarifier also indicated a deep, well-mixed surface layer in the tank (Figures 18A-C). Water temperature values in the surface layer varied only slightly, between 13.6°C and 13.9°C , in both the vertical and radial directions. Unlike the slight indication of a bottom layer inflow in the BENDI primary, the BENDI secondary showed clear evidence of a thin bottom current (Figure 18B). Warmer water temperatures were

Figure 18. BENDI secondary vertical temperature profiles from 1-22-89 (12MN-3AM) at radial distance of A) 13 ft; B) 25 ft; C) 37 ft.

consistently recorded for the deepest thermistor at a depth of 7.5 ft and radial distance of $R = r_{t2} = 25$ ft (Figure 10, Table II). However, as will be discussed in the analysis of the BENDII secondary results, the appearance of the higher water temperature at an intermediate radial distance, rather than at the inlet, is quite surprising.

BENDII

After the BENDI study, the results were reviewed and improvements were made in the measurement techniques. Though inflow density currents in the secondary clarifier were expected due to high suspended solids (SS) concentration (~2000 mg/l), data from the primary clarifier also showed evidence of inflow density currents. A very thin, plunging inflow layer may have passed below the deepest positioned thermistors because of the inflow inertia and the suspended solids concentration of the influent. For this reason, thermistors were positioned deeper in the later studies (Table II).

In the BENDII primary clarifier study, temperatures at most of the 15 sampling points were within 0.3°C of each other (Figures 19A-C). These results are similar to those of the earlier BENDI primary study and indicate well-mixed conditions in the tank. However, a slight temperature difference between the colder top layer and the warmer bottom layer was still apparent at $R=19$ ft and can be seen in Figure 19B.

Figure 19. BENDII primary vertical temperature profiles from 3-4-89 (2AM-6AM) at radial distance of: A) 9 ft; B) 19 ft; C) 30 ft.

The results of the BENDI and BENDII secondary clarifier study are very similar. This is demonstrated by comparing Figures 18A-C with Figures 20A-C. As in the BENDI studies, a deep, well-mixed surface layer, existing over a thin bottom inflow layer, was indicated. Figures 20A-C show that temperatures in the surface layer averaged about 12.6°C at many of the sampling depths. As in the BENDI secondary (Figure 18B), bottom layer temperatures consistently measured warmer ($\sim 13.6^{\circ}\text{C}$) than the surface layer at $r_{t2} = 25$ ft (Figures 20B and 21C). Though the temperature difference of 1°C between the two layers is small, it is still significantly greater than the measurement error ($\pm 0.2^{\circ}\text{C}$). As indicated earlier, greater bottom temperatures are evidence of a plunging inflow layer in the tank.

A comparison of air temperature, tank flow rates and water tank temperatures is provided in Figures 21A-C. Though flow rates declined slightly over the study period from 1.7 MGD to 1.1 MGD, there appeared to be little effect on tank temperature distribution.

One apparent inconsistency, in both the BENDI and BENDII secondary clarifier studies, is evidence of warm temperatures first appearing at intermediate radial distances downstream. It was expected that the thermistor array nearest the inlet baffle, $r_{t1} = 13$ ft, would record the highest temperatures (Figure 10). However, as mentioned previously, the warmest temperatures appeared at $r_{t2} = 25$ ft.

Figure 20. BENDII secondary vertical temperature profiles from 3-5-89 (2:30AM-5:30AM) at radial distance of A) 13 ft; B) 25 ft; C) 37 ft.

Figure 21. BENDII secondary from 3-5-89 (2:30AM-5:30AM): A) air temperatures; B) plant inflow rate; C) vertical water temperatures (R=25 ft).

This can be seen by comparing bottom layer temperatures at $r_{t1} = 13$ ft and $r_{t2} = 25$ ft in Figure 22. At r_{t1} , slightly lower temperatures ($\sim 13.3^{\circ}\text{C}$) were recorded at the deepest thermistor (8.25 ft). This may have occurred because the thermistor was recording temperatures in an interfacial mixing zone between the upper and lower layers.

Suspended solids concentration (SS) had an important affect on the inflow current. Initially, the heavily laden inflow ($C_{SS} \sim 2000$ mg/l) tended to travel along the tank bottom due to the presence of the inlet baffle. At greater radial distances, i.e., those near the outflow weir, some of the SS have settled out. At these distances, the presence of unstable thermal conditions in the water column, i.e., a cold surface layer over a warm inflow layer, would begin to dominate the flow. The warmer temperatures at $r_{t2} = 25$ ft show that the bottom inflow layer is beginning to rise and only mixes with the full depth of the tank beyond this point (Figures 20B-C). Temperatures at the 37 ft radial distance were uniformly low at 12.7°C throughout the water column. This indicated fully-mixed conditions throughout the depth of the tank immediately upstream of the outflow weir. The horizontal profile, presented in Figure 23, shows that temperatures in the surface layer (0.25 ft) were similar to those at the outflow weir.

Note also that Figure 24 indicates that the temperature in the lower layer decreases between $r=25$ ft and $r=37$ ft.

Figure 22. BENDII secondary horizontal temperature profile from 3-5-89 (2:30AM-5:30AM) at a bottom depth of 8.25 ft, showing lower temperature for thermistor closest to inlet.

Figure 23. BENDII secondary horizontal temperature profile from 3-5-89 (2:30AM-5:30AM) at a surface depth of 0.25 ft.

Figure 24. BENDII secondary horizontal temperature profile from 3-5-89 (2:30AM-5:30AM) at a bottom depth of 8.25 ft, showing apparent heat loss in bottom layer.

This occurs probably because of interfacial mixing between the upper and lower flow layers.

OSW

At the Lake Oswego (OSW) site, temperature profiles were recorded over a number of days since the chlorination tanks had no skimmer or rake arms and the thermistors could be left in place. In this way, the longer term effects of a cold weather system on an uncovered treatment tank could be seen.

Throughout the entire study period, which was about 4 days, a definite periodic behavior was seen in the OSW temperature data (Figures 25-27). This periodicity is similar to the diurnal behavior of both the tank inflow rate and ambient air temperatures (Figures 28A-C). After an extended study period, temperatures in the bottom layer began to diverge significantly from those in the surface layer (Figures 27 and 28C). Figure 28C shows clearly that, by the end of the study period, temperatures in the bottom layer are 1°C greater than the surface layer. This temperature difference is maintained even over the diurnal period. Thus, for the time period between 82 and 90 hours (Figure 28C), as temperatures in the bottom layer increase from 11.8°C to 12.6°C , temperatures in the surface layer increase from 10.8°C to 11.8°C .

The divergence of water temperatures between the

Figure 25. OSW chlorination tank vertical temperature profiles from 2-3-89 to 2-7-89 at radial distance 5 ft.

Figure 26. OSW chlorination tank vertical temperature profiles from 2-3-89 to 2-7-89 at radial distance 18 ft.

Figure 27. OSW chlorination tank vertical temperature profiles from 2-3-89 to 2-7-89 at radial distance 33 ft.

Figure 28. OSW chlorination tank from 2-3-89 to 2-7-89: A) air temperatures; B) plant inflow rate; C) vertical water temperatures (P-32 ft).

surface and bottom layer may indicate short circuiting by an underflow in the tank. Temperatures in the bottom layer at $r_{t3}=33$ ft show a definite increase in temperature relative to the rest of the tank. Since $r_{t3}=33$ ft is the position of the thermistor array closest to the outflow weir at $R=37.5$ ft (Figure 10), the inflow layer may remain as a thin underflow across the whole length of the tank. From the standpoint of tank efficiency, this is not desirable. The warm inflow can pass as a bottom layer for the full length of the tank and then, due to thermally induced upward buoyancy or impingement against the tank wall, rise only at the outflow weir. This effectively reduces the tank detention time.

This experimental study has indicated that two layer flow may exist in sedimentation tanks under winter and low flow conditions. The next chapter presents a two layer flow model for circular sedimentation tanks.

CHAPTER VIII

TWO-LAYER FLOW MODEL

MODEL EQUATIONS AND ASSUMPTIONS

The time-averaged, turbulent, fluid mass continuity and momentum equations are presented in cylindrical coordinates in Appendix F. These equations were simplified by making assumptions concerning flow through wastewater tanks. These assumptions are

- 1) axisymmetric flow, i.e., $\bar{v}_\theta = 0$ and $\partial/\partial\theta = 0$;
- 2) the turbulent flux terms ($Re > 10^5$) are very much greater than the molecular terms;
- 3) incompressible flow, i.e., $\partial\bar{\rho}/\partial t = 0$.

Applying the Boussinesq approximation allows for neglecting changes in fluid density except in the body force term. Thus, the mass continuity equation becomes

$$\frac{\partial \bar{w}}{\partial z} + \frac{1}{r} \frac{\partial (r\bar{u})}{\partial r} = 0 \quad (33)$$

The r-momentum equation becomes

$$\begin{aligned} \frac{\partial \bar{u}}{\partial t} + \bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} = & - \frac{1}{\bar{\rho}} \frac{\partial \bar{P}}{\partial r} + \frac{1}{\bar{\rho}} \frac{\partial}{\partial z} (\overline{\rho w' u'}) \\ & + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial r} (r \overline{\rho w' u'}) - \frac{1}{\bar{\rho}} \frac{\partial \bar{P}}{\partial r} = 0 \end{aligned} \quad (34)$$

The z-momentum equation becomes the hydrostatic equation

$$\bar{P} = \bar{P}_a + g \int_z^h \bar{\rho} dz = \bar{P}_a + \bar{\rho}g(h-z) \quad (35)$$

In which $\bar{P}_a$ is the time-averaged atmospheric pressure. All of the terms in equations 33-35 were previously defined in the discussion on hydraulic models (Eqs. 7-9).

Substituting the z-momentum equation (Eq. 35) into the r-momentum equation (Eq. 34) and assuming that $\partial \bar{P}_a / \partial r = 0$, then

$$\begin{aligned} \frac{\partial \bar{u}}{\partial t} + \bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} = & - \frac{g}{\bar{\rho}} \frac{\partial}{\partial r} [\bar{\rho}(h-z)] \\ & + \frac{1}{\bar{\rho}} \frac{\partial}{\partial z} (\overline{\rho w' u'}) + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial r} (r \overline{\rho w' u'}) \end{aligned} \quad (36)$$

The turbulent flux term, $\bar{w}'\bar{u}'$, can be modeled by using the eddy viscosity concept

$$\overline{w' u'} = \nu_t \left(\frac{\partial \bar{u}}{\partial z} + \frac{\partial \bar{w}}{\partial r} \right) \quad (37)$$

If it is assumed that $\bar{w} \ll \bar{u}$ and $\partial \bar{w} / \partial r \ll \partial \bar{u} / \partial z$, the above equation simplifies further

$$\overline{w' u'} \cong \nu_t \left(\frac{\partial \bar{u}}{\partial z} \right) \quad (38)$$

The turbulent stress, τ , which is defined as $\tau = \bar{\rho} \nu_t (\partial \bar{u} / \partial z)$ where $\nu_t = f\{z\}$, can now be substituted into the r-

momentum equation above

$$\frac{\partial \bar{u}}{\partial t} + \bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} = - \frac{g}{\bar{\rho}} \frac{\partial}{\partial r} [\bar{\rho}(h-z)] + \frac{1}{\bar{\rho}} \frac{\partial (\tau\{z\})}{\partial z} \quad (39)$$

Note that the longitudinal turbulent flux term has been neglected in Equation 39 since a simple scaling technique will show that the longitudinal stress, $\tau(r)$, is very much smaller than the vertical stress, $\tau(z)$.

Expressing the continuity and r-momentum equations for each of the two layers (using terms defined in Figure 29), beginning with layer 1 ($h \geq z \geq h_2$)

$$\frac{\partial \bar{w}_1}{\partial z} + \frac{1}{r} \frac{\partial (r \bar{u}_1)}{\partial r} = 0 \quad (40)$$

$$\begin{aligned} \frac{\partial \bar{u}_1}{\partial t} + \bar{u}_1 \frac{\partial \bar{u}_1}{\partial r} + \bar{w}_1 \frac{\partial \bar{u}_1}{\partial z} \\ = - \frac{g}{\bar{\rho}_1} \frac{\partial}{\partial r} [\bar{\rho}_1(h-z)] + \frac{1}{\bar{\rho}_1} \frac{\partial (\tau\{z\})}{\partial z} \end{aligned} \quad (41)$$

For layer 2 ($h_2 \geq z \geq 0$)

$$\frac{\partial \bar{w}_2}{\partial z} + \frac{1}{r} \frac{\partial (r \bar{u}_2)}{\partial r} = 0 \quad (42)$$

$$\begin{aligned} \frac{\partial \bar{u}_2}{\partial t} + \bar{u}_2 \frac{\partial \bar{u}_2}{\partial r} + \bar{w}_2 \frac{\partial \bar{u}_2}{\partial z} \\ = - \frac{g}{\bar{\rho}_2} \frac{\partial}{\partial r} [\bar{\rho}_2(h-z)] + \frac{1}{\bar{\rho}_2} \frac{\partial (\tau\{z\})}{\partial z} \end{aligned} \quad (43)$$

Figure 29. Diagram defining model terms.

DEPTH AVERAGING THE TWO-LAYER CONTINUITY EQUATIONS

For layer 1 ($h \geq z \geq h_2$) assuming no turbulent entrainment or mixing between the layers and averaging over the layer depth (Figure 29), the continuity equation (Eq. 40) becomes

$$\int_{h_2}^h \frac{\partial \bar{w}_1}{\partial z} dz + \int_{h_2}^h \frac{1}{r} \frac{\partial (r \bar{u}_1)}{\partial r} dz = 0 \quad (44)$$

$$\int_{\bar{w}(h_2)}^{\bar{w}(h)} \frac{\partial \bar{w}_1}{\partial z} dz + \frac{1}{r} \int_{h_2}^h \frac{\partial (r \bar{u}_1)}{\partial r} dz = 0 \quad (45)$$

The second term in Equation 45 can be simplified using Liebnitz's rule

$$\int_{z_1}^{z_2} \frac{\partial u}{\partial r} dz = \frac{\partial}{\partial r} \int u dz - u(z_2) \frac{\partial z_2}{\partial r} + u(z_1) \frac{\partial z_1}{\partial r} \quad (46)$$

Thus, the continuity equation (Eq. 45) becomes

$$\left[\bar{w}(h) - \bar{w}(h_2) \right] + \frac{1}{r} \left[\frac{\partial}{\partial r} \int_{h_2}^h (r \bar{u}_1) dz - \frac{\partial h}{\partial r} (r \bar{u}_1(h)) + \frac{\partial h_2}{\partial r} (r \bar{u}_1(h_2)) \right] = 0 \quad (47)$$

Where $\bar{w}(h)$ and $\bar{w}(h_2)$ are the kinematic boundary conditions

$$\bar{w}(h) = \frac{dh}{dt} = \frac{\partial h}{\partial t} + \bar{u}_1(h) \frac{\partial h}{\partial r} \quad (48)$$

$$\bar{w}\{h_2\} = \frac{dh_2}{dt} = \frac{\partial h_2}{\partial t} + \bar{u}_1\{h_2\} \frac{\partial h_2}{\partial r} \quad (49)$$

Substituting these boundary conditions (Eq. 48 and 49) into the continuity equation (Eq. 47), completing the integration and recalling that

$$u_1 h_1 = \int_{h_2}^h \bar{u}_1 dz \quad (50)$$

Where $h_1 = h - h_2$ and u_1 (unbarred) indicates the depth-averaged horizontal velocity (Figure 29). This is done for convenience and distinguishes the time-averaged (overbarred) quantities from the depth-averaged quantities. The continuity equation (Eq. 47) now becomes

$$\begin{aligned} - \frac{\partial h_2}{\partial t} - \bar{u}_1\{h_2\} \frac{\partial h_2}{\partial r} + \frac{\partial h}{\partial t} + \bar{u}_1\{h\} \frac{\partial h}{\partial r} \\ + \frac{1}{r} \frac{\partial}{\partial r} (r u_1 h_1) - \frac{\partial h}{\partial r} \bar{u}_1\{h\} + \frac{\partial h_2}{\partial r} \bar{u}_1\{h_2\} = 0 \end{aligned} \quad (51)$$

Simplifying Equation 51, the continuity equation for layer 1 ($h_1 = h - h_2$) becomes

$$\frac{\partial(h_1)}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} (r u h_1) = 0 \quad (52)$$

The derivation of the layer 2 continuity equation (Eq. 42) is similar

$$\frac{\partial(h_2)}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} (r u h_2) = 0 \quad (53)$$

DEPTH AVERAGING THE TWO-LAYER r-MOMENTUM EQUATIONS

Depth averaging the r-momentum equation for layer 1 (Eq. 41);

$$\begin{aligned} & \frac{1}{h_1} \left[\int_{h_2}^h \frac{\partial \bar{u}_1}{\partial t} dz + \int_{h_2}^h \bar{u}_1 \frac{\partial \bar{u}_1}{\partial r} dz + \int_{h_2}^h \bar{w}_1 \frac{\partial \bar{u}_1}{\partial z} dz \right] \\ &= \frac{1}{h_1} \left[- \int_{h_2}^h \frac{g}{\bar{\rho}_1} \frac{\partial}{\partial r} [\bar{\rho}_1 (h-z)] dz + \int_{h_2}^h \frac{1}{\bar{\rho}_1} \frac{\partial (\tau(z))}{\partial z} dz \right] \quad (54) \end{aligned}$$

Simplifying the right hand side (RHS) of the equation and assuming that the time-averaged density ($\bar{\rho}_1$) is approximately equal to the depth-averaged density (ρ_1) (Figure 29)

$$\bar{\rho}_1 \approx \rho_1 \quad (55)$$

Then

$$\text{RHS} = \frac{-g}{\rho_1 h_1} \left[\int_{h_2}^h \frac{\partial}{\partial r} (\rho_1 h) dz - \int_{h_2}^h \frac{\partial}{\partial r} (\rho_1 z) dz \right] + \frac{1}{\rho_1 h_1} \int_{\tau_i}^{\tau_s} \partial \tau \quad (56)$$

Where τ_s and τ_i are defined as in Figure 29.

Applying Liebnitz's rule and simplifying the RHS of the r-momentum equation (Eq. 56) for layer 1

$$\text{RHS} = \frac{-g}{\rho_1} \left[\rho_1 \frac{\partial}{\partial r} (h_1 + h_2) + \frac{h_1}{2} \frac{\partial \rho_1}{\partial r} \right] + \frac{\tau_s - \tau_i}{\rho_1 h_1} \quad (57)$$

Solving the left hand side of the r-momentum equation

for layer 1 (Eq. 54)

$$\text{LHS} = \frac{1}{h_1} \left[\int_{h_2}^h \frac{\partial \bar{u}_1}{\partial t} dz + \int_{h_2}^h \bar{u}_1 \frac{\partial \bar{u}_1}{\partial r} dz + \int_{h_2}^h \bar{w}_1 \frac{\partial \bar{u}_1}{\partial z} dz \right] \quad (58)$$

Substituting the following equations:

$$\bar{u}_1 \frac{\partial \bar{u}_1}{\partial r} = \frac{\partial \bar{u}_1^2}{\partial r} - \bar{u}_1 \frac{\partial \bar{u}_1}{\partial r} \quad (59)$$

$$\bar{w}_1 \frac{\partial \bar{u}_1}{\partial z} = \frac{\partial \bar{u}_1 \bar{w}_1}{\partial z} - \bar{u}_1 \frac{\partial \bar{w}_1}{\partial z} \quad (60)$$

And from the time averaged continuity equation (Eq. 40)

$$\frac{\partial \bar{u}_1}{\partial r} + \frac{\partial \bar{w}_1}{\partial z} = - \frac{\bar{u}_1}{r} \quad (61)$$

The LHS of Equation 58 becomes

$$\begin{aligned} \text{LHS} = \frac{1}{h_1} & \left[\int_{h_2}^h \frac{\partial \bar{u}_1}{\partial t} dz + \int_{h_2}^h \frac{\partial \bar{u}_1^2}{\partial r} dz \right. \\ & \left. + \int_{h_2}^h \frac{\partial \bar{u}_1 \bar{w}_1}{\partial z} dz + \int_{h_2}^h \frac{\bar{u}_1^2}{r} dz \right] \quad (62) \end{aligned}$$

Since vertically uniform flow is being assumed for each layer, it follows that the momentum correction coefficient is approximately equal to 1 (16), i.e., $u_1 w_1 = (1/h_1) \int \bar{u}_1 \bar{w}_1 dz$ and $u_1 u_1 = (1/h_1) \int \bar{u}_1^2 dz$. Thus, using Liebnitz's rule and simplifying, the LHS becomes

$$\text{LHS} = \frac{1}{h_1} \frac{\partial(u_1 h_1)}{\partial t} + \frac{1}{h_1} \frac{\partial(u_1^2 h_1)}{\partial r} + \frac{u_1^2}{r} \quad (63)$$

Equating the right and left hand sides of the equation for layer 1 (Eqs. 57 and 63), the complete r-momentum equation becomes

$$\begin{aligned} & \frac{1}{h_1} \frac{\partial(u_1 h_1)}{\partial t} + \frac{1}{h_1} \frac{\partial(u_1^2 h_1)}{\partial r} + \frac{u_1^2}{r} \\ &= \frac{-g}{\rho_1} \left[\rho_1 \frac{\partial}{\partial r} (h_1 + h_2) + \frac{h_1}{2} \frac{\partial \rho_1}{\partial r} \right] + \frac{\tau_s - \tau_i}{\rho_1 h_1} \end{aligned} \quad (64)$$

By assuming steady-state conditions, i.e. $\partial/\partial t=0$, and making the substitution, $u_1 = Q_1/(2\pi r h_1)$, the r-momentum equation (Eq. 64) can be simplified

$$\begin{aligned} & \frac{Q_1^2}{4g\pi^2 h_1^2 r^2} \left[\frac{1}{r} + \frac{1}{h_1} \frac{dh_1}{dr} \right] \\ &= \frac{d(h_1+h_2)}{dr} + \frac{h_1}{2\rho_1} \frac{d\rho_1}{dr} + \frac{\tau_i - \tau_s}{\rho_1 h_1 g} \end{aligned} \quad (65)$$

Note that Equation 65 is an ordinary differential equation since all changes in the model variables are in terms of r .

The solution of the r-momentum equation for layer 2 is similar to that carried out above except that the z-momentum equation is slightly different for the lower layer and is

$$\bar{P} = \bar{P}_a + \bar{\rho}_1 g h_1 - \bar{\rho}_2 g (h_2 - z) \quad (66)$$

Making this substitution and carrying out the same steps

as for layer 1, the r-momentum equation for layer 2 (Eq. 43) can be expressed as

$$= \frac{\rho_1}{\rho_2} \frac{dh_1}{dr} + \frac{dh_2}{dr} + \frac{h_1}{\rho_1} \frac{d\rho_1}{dr} + \frac{h_2}{2\rho_2} \frac{d\rho_2}{dr} + \frac{\tau_b - \tau_i}{\rho_2 h_2 g} \quad (67)$$

DEFINING MODEL INPUTS

In order to solve Equations 65 and 67, further assumptions and simplifications are required. Two components which contribute to changes in density in this model are changes in temperature and changes in suspended solids concentration. These two components are assumed to vary linearly with r in each layer. The change in fluid density due to temperature changes can be expressed as $\Delta\rho_T = -\beta\rho\Delta T$ where β is the coefficient of thermal expansion and ΔT is the temperature differential across the layer. The change in fluid density due to suspended solids concentration, $\Delta\rho_{SS}$, can also be determined since the change in suspended solids concentration, ΔC_{SS} , is known from experimental results.

Thus, for each layer

$$\frac{d\rho}{dr} = \left[\frac{-\beta\rho\Delta T + \Delta\rho_{SS}}{\Delta r} \right] \quad (68)$$

The shear stress terms, τ , can be defined as

$$\tau = \rho(f/8)(u)|u| \quad (69)$$

[Note: the absolute value of the velocity is taken to maintain the direction of the shear stress.] Using the

definitions $U_1=Q_1/2\pi rh_1$ and $U_2=Q_2/2\pi rh_2$, the bottom shear stress (τ_b) can be defined as

$$\tau_b = (f_b/8) \frac{\rho_2}{h_2^2 r^2} (Q_2) |Q_2| \quad (70)$$

And the interfacial shear stress (τ_i) can be defined as

$$\tau_i = (f_i/8) \left(\frac{\rho_2 + \rho_1}{2r^2} \right) \left(\frac{Q_1}{h_1} - \frac{Q_2}{h_2} \right) \left| \frac{Q_1}{h_1} - \frac{Q_2}{h_2} \right| \quad (71)$$

In the model, it was assumed that there was no wind and no ice cover, thus, $\tau_s=0$. The friction factor, f , was taken equal to 0.01 for both the interfacial and bottom shear, thus, $f_i = f_b = 0.01$ (24).

The two-layer flow equations (Eqs. 65 and 67) are first-order, non-linear, ordinary differential equations and were solved using a Runge-Kutta method. However, in order to get the equations in a form amenable to a solution by this method, the equations were rearranged. Expressed in general form

$$\frac{dh_1}{dr} = f\{r, h_1, h_2\} \quad (72)$$

$$\frac{dh_2}{dr} = f\{r, h_1, h_2\} \quad (73)$$

Once in this form the two-layer equations were solved by an IBM compatible PC. A copy of the FORTRAN code used in the solution can be seen in Appendix G. The FORTRAN

subroutine DIVPRK, called from the fortran IMSL libraries, carried out the Runge-Kutta-Verner calculations (28). This particular Runge-Kutta solution was a fifth and sixth order method which utilized a global error tolerance. The error tolerance controlled the norm of the local error such that the global error was proportional to 0.0005. A typical output file produced by the code can also be seen in Appendix H.

CHAPTER IX

RESULTS: MODEL SIMULATION

MODEL INPUT CONSTANTS

Model input constants include 1) tank geometry presented in Table I 2) total water temperature change, ΔT , from inlet to outlet, derived from experimental data and listed in Table VII 3) initial flow rates, densities, and other model constants tabulated in Table VIII.

TABLE VII

TOTAL TEMPERATURE CHANGE, FROM INLET TO
OUTLET FOR OSW, BENDI AND BENDII

	<u>OSW</u> CHLORINE CONTACT	<u>BENDI</u> PRIMARY CLARIFIER	<u>BENDI</u> SECONDARY CLARIFIER	<u>BENDII</u> PRIMARY CLARIFIER	<u>BENDII</u> SECONDARY CLARIFIER
$T_{in} (^{\circ}F)$ [$^{\circ}C$]	53.6 [12.0]	58.1 [14.5]	58.5 [14.7]	56.7 [13.7]	56.7 [13.7]
$T_{out} (^{\circ}F)$ [$^{\circ}C$]	51.8 [11.0]	57.7 [14.2]	56.7 [13.7]	55.8 [13.2]	54.5 [12.5]
$\Delta T (^{\circ}F)$ [$^{\circ}C$]	-1.8 [-1.0]	-0.6 [-0.3]	-1.8 [-1.0]	-0.9 [-0.5]	-2.2 [-1.2]
$\Delta \rho_T / \rho_O$	+1.3e-4	+4.8e-5	+1.3e-4	+8.0e-5	+1.4e-4

As mentioned previously, the greatest water temperature changes appeared in the bottom layer for most of the experimental results due to interfacial mixing. This is

particularly true of the results for the BENDII secondary clarifier study presented in Figure 24. Values for total temperature change (ΔT) in the surface layer, were taken from the total range of vertical water temperatures recorded in each of the tanks (Table VII).

TABLE VIII

INITIAL FLOW RATES, DENSITIES AND OTHER MODEL CONSTANTS
FOR THE OSW, BENDI AND BENDII SIMULATIONS

	<u>OSW</u> CHLORINE CONTACT	<u>BENDI</u> PRIMARY CLARIFIER	<u>BENDI</u> SECONDARY CLARIFIER	<u>BENDII</u> PRIMARY CLARIFIER	<u>BENDII</u> SECONDARY CLARIFIER
(lbm/ft ³) ρ_o [g/l]	62.396 [999.46]	62.381 [999.15]	62.424 [999.84]	62.388 [999.26]	62.424 [999.84]
(CFS) Q_o [MGD]	3.63 [2.35]	0.93 [0.60]	1.32 [0.85]	1.93 [1.25]	1.04 [0.68]
(°F ⁻¹) β [°C ⁻¹]	7.0E-5 [1.3E-4]	7.0E-5 [1.3E-4]	7.0E-5 [1.3E-4]	7.0E-5 [1.3E-4]	7.0E-5 [1.3E-4]
h_{1o} (ft)	3.4	1.3	2.0	1.3	2.0
h_{2o} (ft)	6.0	8.0	8.0	8.0	8.0

In the model, all vertical entrance mixing was assumed to occur within the baffle distance, r_b (Figure 30). The initial flow rate, Q_o , was increased by the use of a dilution factor, D_v , which characterized the amount of inlet mixing (Figure 31). Thus, the mixed flow rate, Q_i , can be expressed as

$$Q_i = Q_o(D_v) \quad (74)$$

The direction of the recirculating flow (Q_r), which

Figure 30. Tank radial cross-section showing flow domain.

Figure 31. Tank radial cross-section showing general dimensions and flow parameters.

supplies the dilution water at the inlet, was taken opposite in direction to that of the inflow (Figure 31), i.e.,

$$Q_R = -Q_O(D_V - 1) \quad (75)$$

It is apparent above (Eqs. 74 and 75) that if $D_V = 1.0$, there is no dilution and, therefore, no recirculating flow. If $D_V = 2.0$, then the initial mixed flow rate (Q_i) is twice Q_O and the return flow rate (Q_R) is equal to the initial flow rate (Q_O). Previous studies have shown that a minimum dilution factor is about 1.5 (3,25).

Although the dilution factor (D_V) was used to quantify entrance vertical mixing, no entrainment, or any other mixing process, was modeled within the settling zone between r_b and r_f (Figure 30). All heat loss and settlement of suspended solids were assumed to occur entirely within the settling zone (Figure 30).

Immediately upstream of the outlet weir, a zone of converging flow existed, and the model was not applied to this region (Figure 30). However, the exact upstream distance (r_f) to which this zone dominated was not determined and was arbitrarily taken as the distance to the last thermistor probe (r_{t3}). Thus, in the model, r_f was assumed equal to r_{t3} which was about 3 to 4 ft upstream of the weir for the experimental studies (Figures 10 and 30). In the outlet zone, vertically fully-mixed conditions were assumed to exist. This was reasonable since vertical velocities in the outlet zone are significant (1,4).

For each layer, two possible horizontal flow directions were postulated (Figure 32). When the tank had no baffle, a warm inflow layer (Q_i), containing little or no suspended solids (SS), was considered to flow as a surface layer (Figure 32B). The recirculating flow (Q_r) then traveled back from the the beginning of the outlet zone (r_f) to the end of the inlet mixing zone (r_b). Thus, the clockwise (CW) direction of flow as presented in Figure 32B will be the convention for an inflow with no baffle.

Where the tank had a baffle, as all of the study tanks did, the flow was taken as counter-clockwise (CCW) as in Figure 32A. Thus, the inflow was considered to flow in as a bottom layer while the surface layer provided the recirculating flow.

The assumption that the inflow enters as a bottom layer when the baffle is present was consistent with experimental data. This was true even for warm inflows since the flow inertia and, in some cases, SS concentration dominated the inflow immediately downstream of the baffle. This flow behavior was also in agreement with previous investigators, notably Abdel-Gawad and McCorquodale (18), who showed that the velocity distribution at the inlet of the tank resembled subcritical flow under a submerged sluice gate.

The inflow suspended solids concentration, C_{in} , is the mixed inflow concentration. Changes in fluid density ($\Delta\rho_{SS}$) due to settling of suspended solids was considered to vary

Figure 32. Recirculating flow direction in baffled (CCW) and unbaffled (CW) circular clarifiers showing changes in layer fluid density.

linearly with r (Eq. 68) and to take place within the bottom layer regardless of flow direction, i.e., CW or CCW (Figure 32). This is a reasonable assumption for a bottom layer inflow but less true for a surface layer inflow since SS would settle through the entire depth of the tank.

Model inputs for $\Delta\rho_{ss}$ in Table IX were based on typical wastewater treatment plant concentrations for both the primary and secondary clarifiers (1,2,29). Changes in fluid density based on these typical suspended solids concentrations were determined as illustrated in Appendix I.

TABLE IX
MODEL INPUTS FOR FLOWS AFFECTED BY
SUSPENDED SOLIDS CONCENTRATION

	<u>OSW</u> CHLORINE CONTACT	<u>BENDI</u> PRIMARY CLARIFIER	<u>BENDI</u> SECONDARY CLARIFIER	<u>BENDII</u> PRIMARY CLARIFIER	<u>BENDII</u> SECONDARY CLARIFIER
ΔC (mg/l SS)	0	200	2000	200	2000
$\Delta\rho_{1ss}$ (lbm/ft ³) [mg/l]	0 0	0 0	0 0	0 0	0 0
$\Delta\rho_{2ss}$ (lbm/ft ³) [mg/l]	0	-0.004 [-57.00]	-0.04 [-570.0]	-0.004 [-57.00]	-0.04 [-570.0]

OSW CHLORINATION TANK SIMULATION

Since only the chlorine contact chamber at the Lake Oswego (OSW) site was studied, suspended solids concentrations were neglected in the model simulations.

Thus, changes in density were due only to heat losses from surface cooling.

Bottom Layer Inflow

Inlet conditions at the OSW site included a baffle which produced a bottom inflow and a return surface flow, i.e., counter-clockwise flow (CCW) (see Figure 32A). In the surface layer, a total temperature drop of $\Delta T = 1.8^{\circ}\text{F}$ was determined from the experimental results (OSW) and was taken as typical for cold weather conditions (Table VII). A flow rate of $Q = 3.64$ cfs (Table VIII) was used. Additional model constants for the OSW simulation are shown in Tables VIII and IX.

The results of the initial model run are shown in Figure 33 with $\Delta T = 0^{\circ}\text{F}$, i.e., no density differences between the two layers. This model simulation produced a rapidly decreasing bottom layer inflow depth downstream of the inlet baffle. It also showed little sensitivity to variations in the vertical entrance dilution (D_v).

The general effect of increasing upward buoyancy, by increasing ΔT , was to deepen the bottom inflow layer. Figure 34 shows the effect of thermally induced buoyancy on the bottom layer inflow when $\Delta T = 1.8^{\circ}\text{F}$. The rising bottom layer is more sensitive to changes in D_v than in the previous case (Figure 33). Increasing ΔT to 5.0°F (Figure 35), while holding all other inputs constant, reduced the model's sensitivity to D_v .

Figure 33. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T=0^\circ \text{F}$, $h_{o1}=6 \text{ ft}$ and varying D_v .

Figure 34. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 1.8^\circ \text{F}$, $h_{o1} = 6 \text{ ft}$ and varying D_v .

Figure 35. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 5.0^\circ\text{F}$, $h_{o1} = 6\text{ ft}$ and varying D_v .

The layer densities used in the model simulation presented in Figure 35 are shown in Figure 36. Note that the colder surface temperatures produce an unstable density profile. This is typical for the model when simulating warm inflows affected by the presence of an inlet baffle. The induced CCW flow (Figure 32A) causes an increase in fluid density as the surface layer return flow cools passing from the outlet zone to the inlet baffle (Figure 36).

Varying the flow rate from $Q=3.6$ cfs through $Q=36.0$ cfs while D_v and ΔT were held constant at 1.5 and 0°F , respectively, produced little variation (Figure 37) in the results compared to Figure 33, except at $Q=360$ cfs, where increased flow inertia governed the layer depth profile.

The effect of varying the initial surface layer height, h_{01} , i.e., height of the baffle, illustrates the importance of inlet geometry (Figure 38). With $\Delta T=1.8^\circ\text{F}$, $D_v=1.5$ and other model inputs held constant, the initial surface layer depth (h_{01}) was varied between 2 ft and 7.7 ft. Figure 38 shows that, with decreasing baffle height, the slope of the layer interface becomes more linear. At $h_{01}=2$ ft, the slope of the interface was approximately linear.

Shallower initial surface layer heights also imply that decreasing the depth of the baffle would result in more efficient tank operation by reducing underflow velocities and increasing detention times. Realistically, however, the shallower the depth of the baffle, the more likely thermal short-circuiting could occur.

Figure 36. OSW chlorination tank model simulation:
 plot of density vs. radial distance assuming CCW flow,
 $\Delta T = 5.0^{\circ}\text{F}$ and $D_v = 1.5$.
 v

Figure 37. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 0^\circ\text{F}$ and varying flow rate.

Figure 38. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 1.8^\circ\text{F}$, and varying h_{o1} .

Surface Layer Inflow

In order to simulate tank hydrodynamics without the baffle, the warm inflow layer was considered to enter the tank as a surface layer. This is one of the assumptions made by Jirka (24) in the Pond number analysis and implies stable thermal conditions in the tank, i.e., warm water over cold water. According to the convention in Figure 32B, the direction of the circulating flow was clockwise (CW).

Without the presence of the inlet baffle, determining the initial surface layer height, h_{01} , becomes less straightforward. To demonstrate this, h_{01} was varied while ΔT and D_v were held constant at 1.8°F and 1.5, respectively (Figure 39). Initial surface layer depths, chosen at intermediate tank depths between 3 and 7.7 ft, gave similar results. Immediately downstream of the baffle, there were slight differences in the shape of the layer interfaces. However, at larger radial distances, the effects of the inlet conditions no longer dominated the flow regime and the warm inflowing surface layer became shallower with increasing r . Values of h_{01} chosen near the surface produced significantly different layer interface shapes than for $h_{01} > 2$ ft. As shown in Figure 39, warm surface inflows are capable of short-circuiting across the top of the tank decreasing initial surface layer depth, h_{01} .

Like the previous cases with a bottom inflow (Figures 33, 34 and 36), the surface layer inflow (CW flow)

Figure 39. OSW chlorination tank model simulation:
 plot of layer interface vs. radial distance
 assuming CW flow, $\Delta T = 1.8^\circ\text{F}$, $h_{o1} = 6$ ft and $D_v = 1.5$.

simulation was only slightly sensitive to D_v . At $h_{O1}=6$ ft, the CW flow simulation produced little variation in the surface layer depth (Figure 40). Taking $h_{O1}=3$ ft, demonstrated that, while there was slightly more sensitivity to D_v , the thermally stable surface flow (Figure 41) was less affected by increasing flow inertia than the thermally unstable bottom flow (Figure 33).

BENDI PRIMARY CLARIFIER SIMULATION

The BENDI and BENDII experimental studies showed a deep, well-mixed surface layer in the primary clarifier (Figures 16 and 19). Water temperatures recorded across the primary clarifier are shown in Table VII. Input constants and initial layer depths for the BENDI primary clarifier study are shown in Tables VIII and IX.

Bottom Layer Inflow

A bottom inflow layer was assumed due to the presence of a baffle. When SS were neglected in the simulation, thermally induced buoyancy produced a rising bottom layer (Figure 42). This simulation was slightly sensitive to changing D_v .

Increasing the inflow rate from 0.93 cfs through 930 cfs with $D_v=1.5$ demonstrated the effect of increasing flow inertia on the rising bottom inflow (Figure 43). At greater inflows, i.e., $Q>9.3$ cfs, thermally induced buoyancy had little effect. Note that for $Q=930$ cfs, the velocities

Figure 40. OSW chlorination tank model simulation: plot of layer interface vs. radial distance assuming CW flow, $\Delta T = 1.8^\circ\text{F}$, $h_{o1} = 6\text{ ft}$ and varying D_v .

Figure 42. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 2.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, neglecting SS and varying D_v .

Figure 43. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 2.0$ °F, $h_{o1} = 8$ ft, neglecting SS and varying flow rate.

increase rapidly and the model becomes unstable predicting negative layer depths. Figure 44 shows similar results to those of Figure 42 as ΔT is increased from 2.0°F to 5.0°F .

In cases where typical primary clarifier inflow SS concentrations ($\sim 200 \text{ mg}/\ell$) were included in the simulation, the effect of thermally induced buoyancy was reduced. In Figure 45, the inflow layer can be seen to level off, i.e., $dh/dr \rightarrow 0$, compared to the simulation without SS loading (Figure 42). At $\Delta T = 5.0^{\circ}\text{F}$, the effect of thermally induced buoyancy was much greater than density created differences caused by SS (Figure 46). In this case, the shape of the layer interface approached a vertical asymptote, i.e., $dh/dr \rightarrow +\infty$, seen in the previous OSW and BEND simulations.

Surface Layer Inflow

Without a baffle, the warm inflow was assumed to enter as a surface layer, i.e., CW flow. In this case, the influent cooled ($+\Delta\rho_T$) as it passed from the inlet zone to the outlet zone (Figure 32B). Suspended solids then traveled the full length of the surface layer before being allowed to settle out ($-\Delta\rho_{SS}$) in the return flow.

With an initial depth, h_{o1} , taken as 8.0 ft and $\Delta T = 2.0^{\circ}\text{F}$, the SS ($\sim 200 \text{ mg}/\ell$) laden inflow decreased rapidly with radial distance downstream of the inlet zone (Figure 47). This generated a layer interface similar to that in Figure 45 which was produced by a bottom layer inflow. Although the flow directions in Figures 45 and 47 are

Figure 44. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 5.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, neglecting SS and varying D_v .

Figure 45. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 2.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, $C_{SS} = 200 \text{ mg/l}$ and varying D_v .

Figure 46. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 5.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, $C_{ss} = 200 \text{ mg/l}$ and varying D_v .

Figure 47. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CW flow, $\Delta T = 2.0^\circ\text{F}$, $h_{o1} = 8\text{ ft}$, $C_{ss} = 200\text{ mg/l}$ and $D_v = 1.5$.

opposite, the similarity of the interfaces is due to the density differences between the two layers which, in both simulations, are very small ($\sim 0.003 \text{ lbm/ft}^3$). At $\Delta T = 5.0^\circ\text{F}$, the surface layer depth decreased rapidly with radial distance as it approached the outlet weir, i.e., dh/dr went to $+\infty$ (Figure 48). The stable density distribution produced by these conditions is shown in Figure 49.

Due to the presence of SS, the effect of varying the initial flow depth (h_{o1}) produced slightly different results from those found in the Lake Oswego simulation presented in Figure 39. At $\Delta T = 2^\circ\text{F}$, all of the selected initial surface layer depths converged to $dh/dr = 0$ at $h_1 \approx 3 \text{ ft}$ (Figure 50). At $\Delta T = 5^\circ\text{F}$, the effect of SS on the surface flow was not apparent (Figure 51) and resembled the OSW results in Figure 39.

BENDII SECONDARY CLARIFIER SIMULATION

Bottom Layer Inflow

The CCW model simulation of the BENDII secondary clarifier was the most affected by the presence of SS. Input model constants were shown in Tables VII-IX. An initial inflow SS concentration of 2000 mg/l was used. Although a typical ΔT value, derived from the experimental study, was taken as 2.2°F and was used in the simulation, the model was relatively unaffected by variations in D_v (Figure 52). Figure 53 shows the stable density profile produced by the

Figure 48. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CW flow, $\Delta T = 5.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, $C_{ss} = 200 \text{ mg/l}$ and varying D_v .

Figure 49. BENDI primary clarifier model simulation: plot of density vs. radial distance assuming CW flow, $\Delta T = 5.0^{\circ}\text{F}$, $h_{o1} = 8 \text{ ft}$, $D_v = 1.5$ and $C_{SS} = 200 \text{ mg/l}$.

Figure 50. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CW flow, $\Delta T = 2.0$ °F, $D_v = 1.5$, $C_{ss} = 200$ mg/l, and varying h_{o1} .

Figure 51. BENDI primary clarifier model simulation: plot of layer interface vs. radial distance assuming CW flow, $\Delta T = 5.0$ °F, $D_v = 1.5$, $C_{ss} = 200$ mg/l and varying h_{o1} .

Figure 52. BENDII secondary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 2.2^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, $C_{ss} = 200 \text{ mg/l}$ and varying D_v .

Figure 53. BENDII secondary clarifier model simulation: plot of density vs. radial distance assuming CCW flow, $\Delta T = 2.2^{\circ}\text{F}$, $h_{01} = 8 \text{ ft}$, $C_{ss} = 2000 \text{ mg/l}$ and $D_v = 1.5$.

heavily laden SS inflow even though bottom layer temperatures were greater than surface layer temperatures. The bottom inflow layer was dominated by the presence of SS and its depth, h_2 , remained constant for almost the entire length of the tank. These results are similar to those predicted by DeVantier and Larock's model (1) for secondary clarifiers. Positive, upward buoyancy effects due to ΔT were only noticeable near the outflow zone after the majority of SS had settled out. Increasing ΔT to 5.0°F allows the effect of thermally induced buoyancy to dominate (Figure 54).

As a means of verifying the importance of SS on the secondary clarifier operation, SS were neglected in two model simulations at $\Delta T=2.2^{\circ}\text{F}$ and $\Delta T= 5.0^{\circ}\text{F}$ (Figures 55 and 56). These results were similar to those from the OSW study (Figures 34 and 35) which also neglected SS. Low inflow inertia due to low flow rates (~ 1.0 cfs) and the lack of SS in the model simulation, increased the significance of thermally induced buoyancy. The vertical dilution factor (D_v) was also varied in these simulations but showed no response (Figures 55 and 56). This was unlike the OSW case which was slightly sensitive to changes in D_v (Figures 34 and 35).

Figure 54. BENDII secondary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW Flow, $\Delta T = 5.0^\circ\text{F}$, $h_{o1} = 8 \text{ ft}$, $C_{ss} = 2000 \text{ mg/l}$ and varying D_v .

Figure 55. BENDII secondary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 2.2^\circ\text{F}$, $h_{o1} = 8 \text{ ft}$, neglecting SS and varying D_v .

Figure 56. BENDII secondary clarifier model simulation: plot of layer interface vs. radial distance assuming CCW flow, $\Delta T = 5.0^\circ \text{F}$, $h_{o1} = 8 \text{ ft}$, neglecting SS and varying D_v .

CHAPTER X

SUMMARY AND CONCLUSIONS

This research study has served as a preliminary investigation into density affected, two-layer flow in wastewater treatment tanks under winter conditions. In the experimental studies, higher tank water temperatures were consistently recorded by the deepest placed thermistors suggesting a bottom layer inflow. The presence of the inlet baffle in all of the study tanks caused the inflow to behave as a subcritical flow passing beneath a submerged sluice gate. Thus, even when the inflow was at a greater temperature than the tank, a bottom layer inflow was present. Significant convective turbulence in the surface layer because of surface heat loss was also apparent in the experimental studies and indicated that particle settlement could be adversely affected by surface cooling.

No evidence of short-circuiting of the warm inflow water across the surface of the tank was revealed by the experimental studies. Though rising inflow currents were indicated, these entered the tank as bottom currents and rose only at greater radial distances. Warmer temperatures were not found at the outflow weir, or across the water surface, because surface cooling produced convective mixing

of the surface layer with the warm, rising inflow current. The presence of convective, and/or inflow turbulent mixing would produce a more uniform vertical temperature profile with increasing radial distance. Thus, it was not surprising that outflow temperatures reflected temperatures across the tank surface layer. In addition, many of the temperature profiles, particularly the Bend primary studies, showed fully-mixed vertical conditions immediately upstream of the outlet weir.

The Bend secondary clarifier studies showed a distinct inflow bottom layer at radial distances just upstream of the outflow weir. At large radial distances, the effects of the inlet baffle and suspended solids ceased dominating the flow and thermal instability at or near the outflow weir became significant. This condition may produce short-circuiting of the warm underflow.

Further improvements in the experimental methods used in this study can also be made. Plant-wide temperature data would allow a more detailed investigation of the lag response of the tank water to meteorological conditions. Measuring the sludge layer depth throughout the study period, and at a variety of radial cross-sections, would help validate the assumption of a constant sludge layer surface. The clearance under the inlet baffle, directly affected by the depth of the sludge layer surface, was shown to be an important physical parameter affecting the inflow.

In the model simulation, the sludge layer surface was assumed to be constant both temporally and spatially. However, these assumptions were based on sludge layer surface measurements taken at only one radial cross-section. The rotating rake arms make these assumptions subject to question since evidence was shown that the sludge layer depth was distorted as the rake arms passed. Finally, vertical and horizontal velocity measurements would support conclusions made about the inflow and return flow layers.

From the analysis of the experimental results, a simple numerical model was developed to simulate two-layer flow in circular wastewater treatment tanks. The model equations for each layer (Eqs. 65 and 67) were derived from simplified mass continuity and momentum conservation equations. A FORTRAN code was used to perform the solution of the non-linear differential equations by a Runge-Kutta method.

Model simulations, using input constants derived from the experimental results, were carried out for a variety of conditions. Flexibility in the model allowed the effect of inlet baffle conditions to be investigated in the theoretical study. When the baffle was present, the model predicted the deep, well-mixed surface layer found in the experimental studies. The model also predicted the warm bottom inflow layer induced by the presence of the baffle.

Model simulations assuming no inlet baffle showed that short-circuiting across the surface of the tank would occur.

The surface layer depth and tank detention time were very sensitive to initial inflow depth and inflow density.

Turbulent mixing across the layer interface was not simulated in the model although mixing and entrainment terms would enhance the model's predictive capabilities (29). Such an enhancement would allow the transfer of fluid and solids between the surface layer and the bottom inflow layer.

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APPENDIX A

THERMISTOR COMPONENTS: LINEAR VOLTAGE/TEMPERATURE RELATION

The thermilinear themistor network used in the experimental study was a composite device consisting of resistors and precise thermistors which produced an output voltage linear with temperature. Equations which describe the behavior of the voltage mode are

$$E_{out1} = (-0.0056846)(E_{in})(T) + 0.805858(E_{in}) \quad (76)$$

$$E_{out2} = (+0.0056846)(E_{in})(T) + 0.194142(E_{in}) \quad (77)$$

Where

T = temperature recorded by data logger;

E_{out1} = output voltage of analog channel 1;

E_{out2} = output voltage of analog channel 2.

Since the data logger (Unidata Model 6003A) had an analog voltage input of

$$E_{in} = +5.00 \text{ volts};$$

$$E_{out1} = +2.55 \text{ volts};$$

$$E_{out2} = -2.55 \text{ volts}.$$

Rearranging and solving for T_{max} and T_{min} , which represent the extremes of the temperature scale, then Equation 76 becomes

$$T_{max} = 141.7616 - 175.9139(E_{out1}/E_{in})$$

$$T_{max} = 141.7616 - 175.9139(+2.55/+5.00) = +52.05 \text{ }^{\circ}\text{C} \quad (78)$$

Equation 77 becomes

$$T_{min} = -34.1523 + 175.9139(E_{out2}/E_{in})$$

$$T_{min} = -34.1523 + 175.9139(-2.55/+5.00) = -123.87 \text{ }^{\circ}\text{C} \quad (79)$$

The initial temperature range before calibration is

+52.05°C to -123.87°C. Note that this is not the actual range (-5°C to 45°C) under which the thermistors will function accurately, but only a theoretical range where voltage and temperature are linearly related. Each thermistor was calibrated individually by editing the DCF file (Appendix B).

APPENDIX B

LOGGER SUPPORT SOFTWARE: THERMISTOR
DISPLAY COMMAND FILE (DCF)

```
echo 0
cycle 5
project THM1
source THM1$z
interval 120
entry 1
name Time
formula time
using time
store ud
entry
bytes 0 to 1
name AV A1
signed 13
using ###.##
scale -115.40 to 60.53
units DEG C
store sf
entry
bytes 2 to 3
name AV A2
signed 13
using ###.##
scale -114.83 to 61.09
units DEG C
store sf
entry
bytes 4 to 5
name AV A3
signed 13
using ###.##
scale -114.75 to 61.17
units DEG C
store sf
entry
bytes 6 to 7
name AV A4
signed 13
using ###.##
scale -114.39 to 61.53
units DEG C
store sf
entry
bytes 8 to 9
name AV A5
signed 13
using ###.##
scale -114.02 to 61.90
units DEG C
store sf
entry
```

```
bytes 10 to 11
name AV A6
signed 13
using ###.##
scale -113.61 to 62.31
units DEG C
store sf
entry
bytes 12 to 13
name AV A7
signed 13
using ###.##
scale -113.28 to 62.64
units DEG C
store sf
entry
bytes 14 to 15
name AV A8
signed 13
using ###.##
scale -112.98 to 62.94
units DEG C
store sf
logsize 16
select buffer 0
title Scheme THM1 - , Data from start to end
format "hh:mm","mo/dd/yy"
dump thm1#.prn
go
wait
end
```


APPENDIX C

LOGGER SUPPORT SOFTWARE: MET STATION

DISPLAY COMMAND FILE (DCF)

```
echo 0
cycle 5
project MET
source MET$z
interval 120
entry 1
name Time
formula time
using time
store ud
entry
byte 0
name AV BAR
using ##.##
scale 13.92 to 15.37
units PSI
store sf
entry
byte 1
name AV Speed
using ##.#
scale 0 to 71.304
units mph
store sf
entry
byte 2
name AV Temp
using ###
scale 14 to 140
units DegF
store sf
entry
byte 3
name AV Radn
using ####
formula solar
units W/m2
store sf
entry
byte 4
name AV Dirn
using ###
formula dirn
units Deg
store sf
entry
byte 6
name AV R.H.
using ###
formula humidity
units %
```

```
store sf
logsize 7
select buffer 0
title Scheme MET - , Data from start to end
format hh:mm mo/dd/yy
dump met#.prn
go
wait
end
```

APPENDIX D

DETERMINATION OF THE RICHARDSON FLUX NUMBER AND THE TURBULENT VELOCITY SCALE

Equation 31 defines u_* as

$$u_* = \bar{u} \left(\frac{f_o}{8} \right)^{1/2}$$

Where $\bar{u}$ is the average horizontal velocity for a circular tank and is determined by

$$\bar{u} = \frac{Q}{A\{r\}} = \frac{Q}{2\pi h} \left(\frac{1}{2} \right) \left(\frac{1}{r_i} + \frac{1}{r_f} \right) \quad (80)$$

For the Bend secondary clarifier, from Tables I and IV:

$$Q = 1.04 \text{ cfs};$$

$$h = 10 \text{ ft};$$

$$r_i = r_b = 7 \text{ ft};$$

$$r_f = R = 40 \text{ ft}.$$

Equation 80 then becomes

$$\bar{u} = \frac{1.04}{2\pi(10)} \left(\frac{1}{2} \right) \left(\frac{1}{7} + \frac{1}{40} \right) = 0.0014 \text{ fps} \quad (81)$$

If a typical value for f_o is taken as 0.01, then Equation 31 becomes;

$$u_* = (0.0014)(0.01/8)^{1/2} = 5.0 \times 10^{-5} \text{ fps (0.0015 cm/s)} \quad (82)$$

The buoyancy flux can be determined from Equation 30:

$$R = \frac{-\beta g}{\rho c_v} \phi_n$$

For water, the constants in the above equation are

$$\rho = 0.9997 \text{ g/cm}^3 \text{ at } 10^\circ \text{C};$$

$$c_v \cong c_p = 4.186 \text{ J/g} \cdot ^\circ \text{C};$$

$$\beta = 10^{-4} \text{ } ^\circ \text{C}^{-1};$$

$$g = 981.0 \text{ cm/s}^2.$$

If ϕ_n is in units of W/m^2 , then the buoyancy flux, in cm^2/s^3 , can be derived from the equation above as

$$R = \frac{-(10^{-4})(981)}{(0.997)(4.186)} \phi_n = -2.34 \times 10^{-6} (\phi_n)$$

Using $\phi_n = -84.4 \text{ W/m}^2$ derived from the BENDII secondary clarifier data (Table VI),

$$R = -2.34 \times 10^{-6} (-84.4) = 2.0 \times 10^{-4} \text{ cm}^2/\text{s}^3 \quad (83)$$

The positive value indicates that the direction of the buoyant force is upward.

Once R is determined, the Richardson flux number can be derived by Equation 29,

$$R_f = \frac{-(x)(R)(h)}{u_*^3}$$

If well-mixed conditions exist in the BENDII secondary clarifier, then

$$h = \bar{H}_{\text{eff}} = 10 \text{ ft}.$$

Since it was previously determined in Equations 82 and 83 that

$$R = 2.0 \times 10^{-4} \text{ cm}^2/\text{s}^3,$$

$$u_* = .0015 \text{ cm/s}.$$

With $x \cong 0.4$, Equation 29 becomes

$$R_f = \frac{-(0.4)(2.0 \times 10^{-4})(30.48)(10)}{(0.0015)^3} = -7.2 \times 10^6 \quad (84)$$

Finally, in cases where $R_f \ll -1$, Equation 32 applies and u_t can be estimated as

$$u_t \sim (Rh)^{1/3} = \left| (2.0 \times 10^{-4})(30.48)(10) \right|^{1/3} \quad (85)$$

$$u_t \sim 0.39 \text{ cm/s } (0.013 \text{ fps}) \quad (86)$$

APPENDIX E

DETERMINATION OF THE SURFACE HEAT EXCHANGE COEFFICIENT AND THE RELAXATION TIME

Since ϕ_n and T_s are known from the collected meteorological data, Equation (25) can be used to solve for K. Thus

$$K = \frac{-\phi_n}{T_s - T_E}$$

Using values derived from the BENDII secondary study (Table VI),

$$\phi_n = -84.4 \text{ W/m}^2 ;$$

$$T_E = -8.9 \text{ }^\circ\text{C}.$$

Equation 25 then becomes

$$K = \frac{-(-84.8)}{12.5 - (-8.9)} = 3.96 \frac{\text{W}}{\text{m}^2 \cdot ^\circ\text{C}} \quad (87)$$

The relaxation time, t_r , can be calculated using Equation 28,

$$t_r = \frac{\rho c_p h_s}{K}$$

For water, the constants in the Equation 28 are

$$\rho = 0.9997 \text{ g/cm}^3 \text{ at } 10 \text{ }^\circ\text{C};$$

$$c_p = 4.186 \text{ J/g} \cdot ^\circ\text{C}.$$

If K is in units of $\text{W/m}^2/^\circ\text{C}$ and h_s in ft, then the relaxation time, in days, can be derived from the Equation 28 as

$$t_r = \frac{(0.9997)(4.186)(0.3048)(100)(h_s)}{(K)(10^{-4})(86,400)} = (0.0015) \frac{h_s}{K} \quad (88)$$

Using values from the Bend secondary clarifier and assuming fully-mixed tank conditions, then $h_s = \bar{H}_{eff} = 10$ ft. Thus;

$$t_r = (0.0015) \frac{(10)}{(3.96)} = 37.3 \text{ d} \quad (89)$$

APPENDIX F

TIME-AVERAGED CONSERVATION EQUATIONS

The time-averaged equations describing turbulent flow in wastewater treatment tanks can be expressed in cylindrical coordinates as:

Continuity equation

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{1}{r} \frac{\partial (r \bar{\rho} \bar{u})}{\partial r} + \frac{\partial (\bar{\rho} \bar{w})}{\partial z} + \frac{1}{r} \frac{\partial (\bar{\rho} \bar{v}_{\theta})}{\partial \theta} = 0 \quad (91)$$

Momentum equations in the r, z, and θ directions are

$$\begin{aligned} & \frac{\partial \bar{u}}{\partial t} + \bar{u} \frac{\partial \bar{u}}{\partial r} + \bar{w} \frac{\partial \bar{u}}{\partial z} + \frac{\bar{v}_{\theta}}{r} \frac{\partial \bar{u}}{\partial \theta} - \frac{\bar{v}_{\theta}^2}{r} \\ &= - \frac{1}{\bar{\rho}} \frac{\partial \bar{P}}{\partial r} + g_r + \frac{1}{\bar{\rho}} \frac{\partial}{\partial z} \left(\mu \frac{\partial \bar{u}}{\partial z} - \overline{\rho w' u'} \right) + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial \bar{u}}{\partial r} - r \overline{\rho u'^2} \right) \\ & - \left(\mu \frac{\bar{u}}{r^2} - \frac{\overline{v_{\theta}'^2}}{r} \right) + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial \theta} \left(\frac{\mu}{r} \frac{\partial \bar{u}}{\partial \theta} - \overline{\rho u' v_{\theta}'} \right) - \frac{\mu}{\bar{\rho}} \frac{2}{r^2} \frac{\partial \bar{v}_{\theta}}{\partial \theta} \end{aligned} \quad (92)$$

The z-momentum equation is

$$\begin{aligned} & \frac{\partial \bar{w}}{\partial t} + \bar{u} \frac{\partial \bar{w}}{\partial r} + \bar{w} \frac{\partial \bar{w}}{\partial z} + \frac{\bar{v}_{\theta}}{r} \frac{\partial \bar{w}}{\partial \theta} \\ &= - \frac{1}{\bar{\rho}} \frac{\partial \bar{P}}{\partial z} + \frac{1}{\bar{\rho}} \frac{\partial}{\partial z} \left(\mu \frac{\partial \bar{w}}{\partial z} - \overline{\rho w'^2} \right) + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial \bar{w}}{\partial r} - \overline{\rho r u' w'} \right) \\ & + g_z + \frac{1}{\bar{\rho} r} \frac{\partial}{\partial \theta} \left(\frac{\mu}{r} \frac{\partial \bar{w}}{\partial \theta} - \overline{\rho w' v_{\theta}'} \right) \end{aligned} \quad (93)$$

The θ -momentum equation is

$$\begin{aligned}
 & \frac{\partial \bar{v}_\theta}{\partial t} + \bar{u} \frac{\partial \bar{v}_\theta}{\partial r} + \bar{w} \frac{\partial \bar{v}_\theta}{\partial z} + \frac{\bar{v}_\theta}{r} \frac{\partial \bar{v}_\theta}{\partial \theta} + \frac{\overline{u'v_{\theta'}}}{r} \\
 &= - \frac{1}{\bar{\rho}} \frac{\partial \bar{P}}{\partial \theta} + \frac{1}{\bar{\rho}} \frac{\partial}{\partial z} \left(\mu \frac{\partial \bar{v}_\theta}{\partial z} - \rho \overline{w'v_{\theta'}} \right) + \frac{1}{\bar{\rho}r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial \bar{v}_\theta}{\partial r} - \rho \overline{v_{\theta'}u'} \right) \\
 &- \left(\mu \frac{\bar{v}_\theta}{r^2} + \frac{\overline{u'v_{\theta'}}}{r} \right) + \frac{1}{\bar{\rho}r} \frac{\partial}{\partial \theta} \left(\frac{\mu}{r} \frac{\partial \bar{v}_\theta}{\partial \theta} + 2\mu \frac{\bar{u}}{r} - \rho \overline{v_{\theta'}^2} \right) \quad (94)
 \end{aligned}$$

APPENDIX G

TWO-LAYER MODEL: FORTRAN CODE

\$debug

```

*****
C*TWO LAYER FLOW PROGRAM FOR SEDIMENTATION TANKS- TANK.FOR *
*****
C
C THIS PROGRAM UTILIZES THE RUNGE-KUTTA METHOD TO SOLVE THE
C FIRST ORDER, NON-LINEAR DIFFERENTIAL EQUATIONS RELATING TO
C THE TWO-LAYER FLOW CONDITIONS ASSUMED TO EXIST IN
C SEDIMENTATION TANKS.
C THE GOVERNING EQUATIONS FOR EACH LAYER ARE DERIVED FROM
C THE CONTINUITY, Z AND R-MOMENTUM EQUATIONS. CALCULATIONS
C FOR THESE EQUATIONS ARE CARRIED OUT BY THE SUBROUTINE
C "FCN" WHICH IS CALLED FROM WITHIN THE SUBROUTINE "DIVPRK".
C THIS FORTRAN SUBROUTINE IS PART OF THE IMSL LIBRARIES AND
C REQUIRES THAT THE COMPILED CODE BE LINKED TO THE FOLLOWING
C LIBRARIES; BLAS MATHCORE MATH5. WHEN LINKING THE COMPILED
C CODE TO THESE LIBRARIES, NOTE THAT SPACES ARE THE ONLY
C DELIMITERS RECOGNIZED BY THE MICROSOFT (MS) COMPILER AND
C SHOULD BE ENTERED, EXACTLY AS ABOVE, AT THE APPROPRIATE
C LIBRARY LINKING PROMPT.
C
*****
C*          THIS IS THE MAIN PROGRAM          *
*****
C
      IMPLICIT DOUBLE PRECISION (A-H,O-Z)
      PARAMETER      (MXPARM=50, NEQ=2)
      INTEGER IDO
      DIMENSION PARAM(MXPARM), H(NEQ)
      EXTERNAL FCN, DIVPRK, DSET
      COMMON /DATA/ RT,HT,RI,IRI,N,TOL,RHO1,RHO2,DRS
      N = 0
C
C  PARAM  = VECTOR CONTAINING OPTIONAL PARAMETERS
C  NEQ    = NUMBER OF EQUATIONS TO BE SOLVED
C  FCN    = USER SUPPLIED FUNCTION, IN THIS CASE, DEFINING
C           THE TWO LAYER FLOW EQUATIONS
C  DIVPRK = FORTRAN SUBROUTINE FROM IMSL LIBRARIES UTILIZING
C           RUNGE-KUTTA METHOD FOR SOLVING FIRST-ORDER,
C           NON-LINEAR, ORDINARY DIFFERENTIAL EQUATIONS.
C  DSET   = FORTRAN SUBROUTINE WHICH SETS PARAMETER VECTOR
C           (PARAM) TO ZERO.
C
C
C      OPEN INPUT FILE AND SET INITIAL CONDITIONS
C  RI= INITIAL RADIUS (FT), AT BAFFLE, AT WHICH TWO LAYER
C  FLOW ASSUMED
C  RT= TOTAL RADIUS (FT), FROM THE BAFFLE (RI) TO THE
C  BEGINNING
C  OF THE WITHDRAWAL ZONE
C  HT= TOTAL HEIGHT OF FLOW (FT)

```

```

C      H(1)= HEIGHT (FT) OF SURFACE LAYER,
C          INITIALLY TAKEN AS HEIGHT OF BAFFLE
C      H(2)= INITIAL HEIGHT (FT) OF BOTTOM LAYER, INITIALLY
C          TAKEN AS TOTAL HEIGHT OF FLOW MINUS HEIGHT OF BAFFLE
C
C      READ INITIAL VALUES
C
C      OPEN(UNIT=2,FILE='IN.DAT',STATUS='OLD')
C      READ(2,*)RI,RT,HT,H(1)
C      H(2)= HT-H(1)
C
C          READ ERROR TOLERANCE
C      READ(2,*)TOL
C
C          SET PARAM TO DEFAULT
C      CALL DSET (MXPARM, 0.0, PARAM, 1)
C          SELECT ABSOLUTE ERROR CONTROL
C      PARAM(10) = 1.0
C
C
C      THIS SECTION BEGINS THE LOOPING SEQUENCE WHICH CALLS THE
C      IMSL SUBROUTINE "DIVPRK" THAT UTILIZES THE RUNGE-KUTTA
C      METHOD TO SOLVE THE TWO LAYER FLOW EQUATIONS SPECIFIED IN
C      THE USER SUPPLIED SUBROUTINE "FCN".
C      IDO = FLAG INDICATING STATE OF COMPUTATION, I.E.
C      IDO=1
C      INDICATES INITIAL ENTRY AND IDO=3 INDICATES FINAL
C          CALL TO RELEASE WORKSPACE.
C      REND = RADIAL DISTANCE AT CURRENT STEP
C      DRS = CURRENT CHANGE IN RADIAL DISTANCE (FT)
C      IDO= 1
C      IRI= INT(RI)
C      IRT= INT(RT)
C      DO 10 IRS= IRI,IRT
C          REND = FLOAT(IRS)
C          DRS = RT-REND
C          CALL DIVPRK (IDO, NEQ, FCN, RI, REND, TOL, PARAM, H)
C          WRITE (1,'(I6,4F12.3)') IRS,H(1),RHO1,H(2),RHO2
C          WRITE (5,'(I6,F12.3)')IRS,H(1)*(-1)
10 CONTINUE
C
C      FINAL CALL TO RELEASE WORKSPACE
C
C      IDO = 3
C      CALL DIVPRK (IDO, NEQ, FCN, RI, REND, TOL, PARAM, H)
C      END
C
C*****
C*      TWO LAYER FLOW EQUATIONS (IN ENGLISH UNITS)      *
C*****
C
C      SUBROUTINE FCN (NEQ, REND, H, HPRIME)
C      IMPLICIT DOUBLE PRECISION (A-H,O-Z)
C      INTEGER NEQ

```



```

DIMENSION H(NEQ), HPRIME(NEQ)
COMMON /DATA/ RT,HT,RI,IRI,N,TOL,RHO1,RHO2,DRS
N = N+1
IF(N.NE.1) GOTO 50

```

```

C
C READ IN SYSTEM CONSTANTS
C

```

```

READ(2,*)DV,Q,BETA

```

```

C DV = DILUTION FACTOR (NON-DIMENSIONAL)
C Q = FLOW RATE (CFS): POSITIVE INDICATES COUNTERCLOCKWISE
C FLOW AND NEGATIVE INDICATES CLOCKWISE FLOW.
C

```

```

C Q1      = FLOW (CFS) THROUGH LAYER 1
C Q2      = FLOW (CFS) THROUGH LAYER 2
C DR      = TOTAL CHANGE IN RADIAL DISTANCE (FT)
C DRHO1   = CHANGE IN DENSITY (LBM/FT^3) THROUGH
C          LAYER 1 UP TO CURRENT STEP
C RHO1    = DENSITY (LBM/FT^3) IN LAYER 1 AT CURRENT STEP
C DRHO2   = CHANGE IN DENSITY (LBM/FT^3) THROUGH
C          LAYER 2 UP TO CURRENT STEP
C RHO2    = DENSITY (LBM/FT^3) IN LAYER 2 AT CURRENT STEP
C RHOAVG  = AVERAGE DENSITY (LBM/FT^3) OF LAYERS 1 & 2
C          FOR CALCULATION OF INTERFACIAL SHEAR STRESS
C          AT CURRENT STEP
C BETA     = COEFFICIENT OF THERMAL EXPANSION (F^-1)

```

```

C READ IN INITIAL VALUES FOR LAYER 1
C

```

```

READ(2,*)DT1,RHO1SS

```

```

C DT1     = CHANGE IN TEMPERATURE (F) THROUGH LAYER 1
C RHO1SS  = CHANGE IN DENSITY (LBM/FT^3) DUE TO
C          SETTLING OF SUSPENDED SOLIDS IN LAYER 1
C

```

```

C READ IN INITIAL VALUES FOR LAYER 2
C

```

```

READ(2,*)DT2,RHO2O,RHO2SS

```

```

C DT2     = CHANGE IN TEMPERATURE (F) THROUGH LAYER 2
C RHO2O   = INITIAL DENSITY (LBM/FT^3) IN LAYER 2
C RHO2SS  = CHANGE IN DENSITY (LBM/FT^3) DUE TO
C          SETTLING OF SUSPENDED SOLIDS IN LAYER 2
C

```

```

C BASED ON THE ABOVE INITIAL VALUES, DETERMINE THE REST OF
C THE SYSTEM CONSTANTS
C

```

```

Q1      = -Q*(DV-1.)
Q2      = DV*Q
DR      = RT-RI
DRHO1   = ((-BETA*DT1*RHO2O)+RHO1SS)/DR
DRHO2   = ((-BETA*DT2*RHO2O)+RHO2SS)/DR

```

```
C
C                                     PRINT HEADER
C
OPEN(UNIT=1,FILE='OUT.DAT',STATUS='NEW')
OPEN(UNIT=5,FILE='GOUT.DAT',STATUS='NEW')
WRITE(1,252)
252 FORMAT(15X,'INPUT:',/)
WRITE(1,260)RI
260 FORMAT(2X,'INITIAL RADIUS (FT):',2X,'RI=',1X,F4.1)
WRITE(1,262)RT
262 FORMAT(2X,'TOTAL RADIUS (FT)   : ',2X,'RT=',1X,F4.1)
WRITE(1,264)HT
264 FORMAT(2X,'TOTAL DEPTH (FT)    : ',2X,'HT=',1X,F4.1)
WRITE(1,270)DV
270 FORMAT(2X,'DILUTION FACTOR     : ',2X,'DV=',2X,F5.2)
WRITE(1,272)Q
272 FORMAT(2X,'FLOW RATE (CFS)      : ',2X,'Q=',2X,F5.2)
WRITE(1,268)TOL
268 FORMAT(2X,'ERROR TOLERANCE      : ',2X,'TOL=',2X,F6.4)
WRITE(1,266)H(1)
266 FORMAT(2X,'INITIAL DEPTH OF SURFACE LAYER
1 (FT): ',2X,'H1=',F4.1)
WRITE(1,274)BETA
274 FORMAT(2X,'COEFF. OF THERMAL EXPANSION (F^-1): ',2X,
1 'BETA=',2X,E8.2)
WRITE(1,276)DT1
276 FORMAT(2X,'LAYER 1 CHANGE IN TEMPERATURE
1 (F): ',2X,'DT1=',2X,F5.1)
WRITE(1,282)DT2
282 FORMAT(2X,'LAYER 2 CHANGE IN TEMPERATURE
1 (F): ',2X,'DT2=',2X,F5.1)
WRITE(1,284)RHO2O
284 FORMAT(2X,'LAYER 2 INITIAL DENSITY (LBM/FT^3): ',2X,
1 'RHO2O=',2X,F8.3)
WRITE(1,280)RHO1SS
280 FORMAT(2X,'LAYER 1 CHANGE IN FLUID DENSITY DUE TO SS
1 (LBM/FT^3): ',
1 2X,'RHO1SS=',2X,F5.3)
WRITE(1,286)RHO2SS
286 FORMAT(2X,'LAYER 2 CHANGE IN FLUID DENSITY DUE TO SS
1 (LBM/FT^3): ',
1 2X,'RHO2SS=',2X,F5.3,///)
WRITE(1,288)
288 FORMAT(15X,'OUTPUT:',/)
WRITE(1,300)
300 FORMAT(4X,'r',9X,'h1',9X,'RHO1',9X,'h2',9X,'RHO2',/)
C
C
50  RHO1 = RHO2O+(DRHO2*DR)+(DRHO1*DRS)
    RHO2 = RHO2O+(DRHO2*(DR-DRS))
    RHOAVG = (RHO1 + RHO2)/2
```

DEBUGGING OUTPUT FILE

```

OPEN(UNIT=4, FILE='BUG.DAT', STATUS='NEW')
WRITE(4, 100) REND, H(1), H(2), DRHO1, DRHO2, HPRIME(1),
1 HPRIME(2)
100 format(1x, 'r(ft)=', f10.3, 2x, 'h1(r)=', f10.3, 2x,
1 'h2(r)=', f10.3, 2x, 'drrho1=', e15.5, 2x, 'drrho2=', 2x,
1 e15.5, 2x, 'HPRIME(1)=', 2x, e15.5, 2x, 'HPRIME(2)=',
1 2x, e15.5)

```

SOLVING TWO-LAYER FLOW EQUATIONS

LAYER 1 TERMS

```

A1 = -7.867E-4*(Q1**2)/((H(1)**2)*(REND**3))
B1 = (H(1)/(2.0*RHO1))*DRHO1
QH = Q1/H(1)-Q2/H(2)
C1 = 3.166E-5*(RHOAVG/((REND**2)*RHO1*H(1)))*QH*ABS(QH)
D1 = 7.867E-4*(Q2**2)/((H(2)**2)*(REND**3))
E1 = (-H(1)/RHO2)*DRHO1
DTAUB = (RHO2/((H(2)**2)*(REND**2)))*Q2*ABS(Q2)
DTAUI = (RHOAVG/(REND**2))*QH*ABS(QH)
F1 = (-9.833E-7/(RHO2*H(2)))*(DTAUB - DTAUI)
G1 = (-H(2)/(2.*RHO2))*DRHO2
S1 = 1.-7.867E-4*(Q2**2)/((H(2)**3)*(REND**2))
T1 = 7.867E-4*(Q1**2)/((H(1)**3)*(REND**2))
U1 = ((RHO1/RHO2)/S1)-1.
HPRIME(1) = (A1+B1+C1+((D1+E1+F1+G1)/S1))/(T1+U1)

```

LAYER 2 TERMS

```

A2 = -D1
B2 = 2.0*B1
C2 = -G1
D2 = -F1
E2 = -A1
F2 = -B1
G2 = -C1
S2 = 1.-T1
T2 = 1.-S1
U2 = (RHO1/RHO2)/S2-1.
HPRIME(2) = (A2+B2+C2+D2+((RHO1/RHO2)*(E2+F2+G2)/S2))/
1 (T2+U2)
RETURN

```

END

APPENDIX H

TWO-LAYER MODEL: OUTPUT FILE

INPUT:

INITIAL RADIUS (FT): RI= 7.0
 TOTAL RADIUS (FT) : RT= 37.0
 TOTAL DEPTH (FT) : HT= 10.0
 DILUTION FACTOR : DV= 1.50
 FLOW RATE (CFS) : Q= 1.04
 ERROR TOLERANCE : TOL= .0005
 INITIAL DEPTH OF SURFACE LAYER (FT): H1= 8.0
 COEFF. OF THERMAL EXPANSION (F^{-1}): BETA= .70E-04
 LAYER 1 CHANGE IN TEMPERATURE (F): DT1= -.4
 LAYER 2 CHANGE IN TEMPERATURE (F): DT2= -1.8
 LAYER 2 INITIAL DENSITY (LBM/FT³): RHO20= 62.424
 LAYER 1 CHANGE IN FLUID DENSITY DUE TO SS (LBM/FT³):
 RHO1SS= .000
 LAYER 2 CHANGE IN FLUID DENSITY DUE TO SS (LBM/FT³):
 RHO2SS= -.040

OUTPUT:

r	h1	RHO1	h2	RHO2
7	8.000	62.394	2.000	62.424
8	7.963	62.394	2.037	62.423
9	7.926	62.393	2.074	62.422
10	7.890	62.393	2.110	62.421
11	7.852	62.393	2.148	62.420
12	7.812	62.393	2.188	62.419
13	7.771	62.393	2.229	62.418
14	7.727	62.393	2.272	62.417
15	7.681	62.393	2.319	62.415
16	7.631	62.393	2.369	62.414
17	7.578	62.393	2.422	62.413
18	7.520	62.393	2.479	62.412
19	7.458	62.393	2.542	62.411
20	7.390	62.393	2.610	62.410
21	7.315	62.393	2.685	62.409
22	7.232	62.393	2.768	62.408
23	7.140	62.393	2.860	62.407
24	7.037	62.393	2.963	62.406
25	6.920	62.393	3.079	62.405
26	6.787	62.393	3.213	62.404
27	6.632	62.392	3.368	62.403
28	6.449	62.392	3.551	62.402
29	6.230	62.392	3.770	62.400
30	5.960	62.392	4.040	62.399
31	5.616	62.392	4.383	62.398
32	5.160	62.392	4.840	62.397
33	4.513	62.392	5.487	62.396
34	3.497	62.392	6.503	62.395
35	1.571	62.392	8.429	62.394

APPENDIX I

DETERMINATION OF $\Delta\rho_{\text{SS}}$

Determine the mass of solids, m_{SS} , in 1 liter of total volume, V_T , if $C_{in} = 2000.0 \text{ mg/l}$ (typical for secondary clarifiers). Then

$$m_{SS} = C_{in} V_T = (2000.0 \text{ mg/l})(1 \text{ l}) = 2000.0 \text{ mg} = 2.0000 \text{ g} \quad (95)$$

The density of the suspended solid can be found since $\rho_{SS} = (SG_{SS})(\rho_w)$. Where SG_{SS} is the specific gravity of the suspended solids taken equal to about 1.4 for both the primary and secondary clarifiers (30), and ρ_w is the density of water equal to 1.0000 g/cm^3 , ρ_{SS} becomes

$$\rho_{SS} = (1.4000)(1.0000) = 1.4000 \text{ g/cm}^3 \quad (96)$$

Now, the volume of SS, V_{SS} , can be determined from

$$V_{SS} = m_{SS} / \rho_{SS} = 2.0000 / 1.4000 = 1.4285 \text{ cm}^3 \quad (97)$$

The volume of water, V_w , then becomes

$$V_w = V_T - V_{SS} = 1000.0 - 1.4285 = 998.5915 \text{ cm}^3 \quad (98)$$

The inflow density, ρ_o , can be expressed as

$$\rho_o = \Sigma m_i / V_T = [(998.5915)(1.0000) + (1.4285)(1.4000)] / 1 \text{ l} \quad (99)$$

$$\rho_o = 1000.5714 \text{ g/l}$$

$$\Delta \rho_{SS} = \rho_o - \rho_f = 1000.5714 - 1000.0000 \quad (100)$$

$$\Delta \rho_{SS} = 0.5714 \text{ g/l} = 571.4 \text{ mg/l} = 0.036 \text{ lbm/ft}^3 \quad (101)$$

Note that ρ_f is assumed to be the density of pure water at a specific temperature and implies that all of the SS are settled out as the fluid travels the length of the settling zone.